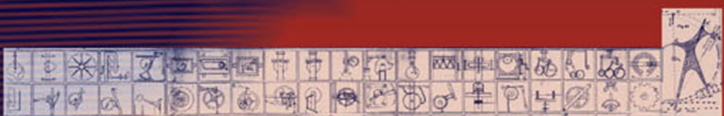
History of Mechanism and Machine Science 5

A. Golovin
V. Tarabarin

Russian Models from the Mechanisms Collection of Bauman University



Springer

Russian Models from the Mechanisms Collection of Bauman University

HISTORY OF MECHANISM AND MACHINE SCIENCE
Volume 5

Series Editor

MARCO CECCARELLI

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This book series aims to establish a well defined forum for Monographs and Proceedings on the History of Mechanism and Machine Science (MMS). The series publishes works that give an overview of the historical developments, from the earliest times up to and including the recent past, of MMS in all its technical aspects.

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Russian Models from the Mechanisms Collection of Bauman University

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ANNOTATION

Bauman University (Imperial Moscow Technical School – Moscow State Technical University) is one of the oldest engineering universities in the world. It is considered to have been established in 1830. The Theory of Mechanisms and Machines Department (TMM) of Bauman University enjoys a mechanisms collection of more than 600 items. Models from collections of F. Redtenbacher, F. Reuleaux and A. Clair are part of the collection. These models are known in the scientific and engineering world, but a large part of the range are mechanisms designed by Russian and Soviet scientists and engineers. Unfortunately, some Russian models of the collection were lost, broken or transferred to other institutions. The "Russian models" collection consists of:

- Original creations of TMM department scientists and postgraduates
- Original creations of Russian and Soviet engineers and scientists
- Models reconstructed by our foremen according to foreign scientists' projects
- Copies of interesting models

Some models were created for pure scientific' interest, other ones were converted into engineering projects and used in real devices. We don't know exactly when the collection was set up. Today its oldest item dates back to 1861. Naturally, while on the subject of the collection aged 150, it would make sense to introduce the history of Bauman University and the TMM Department (earlier Applied Mechanics Department), outline some reasons for the creation of the collection, TMM course contents and mention scientific and pedagogic schools and their founders.

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PREFACE BY THE SERIES EDITOR,
Professor M. Ceccarelli

This book is part of a book series on the History of Mechanism and Machine Science (HMMS).
This series is novel in its concept of treating historical developments with a technical approach to illustrate the evolution of matters of Mechanical Engineering that are related specifically to mechanism and machine science. Thus, books in the series will describe historical developments by mainly looking at technical details with the aim to give interpretations and insights of past achievements. The attention to technical details is used not only to track the past by giving credit to past efforts and solutions but mainly to learn from the past approaches and procedures that can still be of current interest and use both for teaching and research.
The intended re-interpretation and re-formulation of past studies on machines and mechanisms requires technical expertise more than a merely historical perspective, therefore, the books of the series can be characterized by this emphasis on technical information, although historical development will not be overlooked.
Furthermore, the series will offer the possibility of publishing translations of works not originally written in English, and of reprinting works of historical interest that have gone out of print but are currently of interest again.
I believe that the works published in this series will be of interest to a wide range of readers from professionals to students, and from historians to technical researchers. They will all obtain both satisfaction from and motivation for their work by becoming aware of the historical framework which forms the background of their research.
I would like to take this opportunity to thank the authors and editors of these volumes very much for their efforts and the time they have spent in order to share their accumulated information and understanding of the use of past techniques in the history of mechanism and machine science.

Marco Ceccarelli (Chair of the Scientific Editorial Board)
Cassino, April 2007

PREFACE

In 1998 the chairman of the Russian National Committee of TMM Professor Arcady Bessonov, recommended one of authors of this book to be come a member of the IFToMM Permanent Commission on the History of Mechanisms and Machines Sciences (PC HMMS). Willy-nilly from this time the history of technique, as hobby passed on to a serious the employment in the history of engineering science. Interest history of a subject is natural for Professor, a leading a course of Theory of Mechanisms and Machines in Bauman University. This interest is supported by the fact that Bauman University is one of the oldest technical universities in Russia, and the course "Applied Mechanics" – later "Theory of Mechanisms and Machines" was the first systematic course in Russia. The second author supervises a cycle of laboratory works on TMM. Models of mechanisms are placed in laboratory in show-windows of ancient cases quite possibly coevals of the first course. He became interested in contents of these cases: firstly in models, and then in their origin. Later he occupied himself with the creation of a web-site "The Collection of mechanisms in department TMM in Bauman University". Gradually both authors had the idea of cooperation, although several years previously, we could not imagine this happening.
We took an active part in the work of PC HMMS from 2000. It was promoted by of chairman of the commission Professor Marco Ceccarelli. From 2000 to 2006, we participated in two symposiums on the History of MMS (Italy, 2000, 2004) and three Workshops (Germany, Dresden, 2004; Russia, Moscow, 2005; USA, Ithaca, 2006). In 2002, Bauman University's TMM Department was visited by a member of the commission, Professor Teun Koetsier (Netherlands) and in 2003 by Professor Marco Ceccarelli (Italy). Not only lectures to students and professors of did they give University, but were also familiarized with a collection of mechanisms. Since 2000 we have cooperated with professors of a history of mathematics and mechanics cabinet of the M. Lomonosov Moscow State University I. Tiulina and V. Chineneva. Our university is connected with Moscow University, including the general history: in the 19th century and further in the beginning of the 20th century, the majority of Professors read theoretical course, were graduates of Moscow University. A. Letnikov lectured mathematics, N. Joukovsky – theoretical and analytical mechanics, A. Yershov, F. Orlov and N. Mertsalov – applied mechanics. The list can be continued. In Kursk State Technical University, under of direction the Professor S. Jatsun (a member of PC HMMS) conferences on history of a science and techniques were prepared and lead in 2002 and 2003 with our participation. Professors Teun Koetsier (Netherlands) and Marco Ceccarelli (Italy) participated in their work. Since 2001 the section "History of Mechanics and Techniques" is included in the program of international biennial "Vibrating machines and technology", held by Kursk State Technical University. The conference 100 years of Bennet's mechanism in Kazan's State Agricultural Academy was prepared and lead with our participation in 2003. Participation in conferences and the visits of Professors Teun Koetsier and Marco Ceccarelli allowed us to understand that our collection of mechanisms is a treasure. Marco's enthusiasm and literally energy compelled us to flippantly offer Moscow and Department TMM as a place for holding the third Workshop (1st – Austria, 2002; 2nd – Germany, 2004). It is senseless to tell

about the difficulties connected with the organization of the conference. In many respects its success was defined by the financial support of the president of the firm "Rimison" Ph. D. Ekaterina Morozova. It is enough to say, that it was the first International Conference lead by the department. Twenty-four reports of scientists from 12 countries were presented at the conference. Participants were familiarized with our collection and we were once again convinced its uniqueness.

Professor Marco Ceccarelli suggested write book on that we history of engineering science in Bauman University for the publishing house Springer. We supposed that the most suitable is the theme connected with a collection. The TMM Department of Bauman University enjoys a mechanisms collection of more than 600 exhibits. Models from the collections of F. Redtenbacher, F. Reuleaux and A. Clair are part of the collection. These models are known in the scientific and engineering world. We do not know the exact the date of birth of the collection. Today Its oldest item dates back to 1861. But a large part of the range are mechanisms created of Russian and Soviet scientist and engineers. This part of the collection is virtually unknown. Our book partly fills this white space. Partly unknown because for a long time many models was lost or destroyed or farmed out to other institution. Some models were created for pure scientific interest, others were converted in engineering projects and used in real devices. Many of the exhibits are unknown, not only outside Russia, but also outside Bauman University. So the idea to write a work on Russian models of a collection was born. First it seemed to us that we would easily perform this work: here are mechanisms, here is a digital camera, the work of mechanisms is clear, descriptions exist to many models… However, from the first days we understood that it is not simple. Firstly, the sequence of statements is not clear. Certainly, the basic part of the book is the description of mechanisms. A basis of a description should be classification. Which? It is possible to offer classification by years, authors, structure, destination, etc. We decided that "Russian models…" consist of:

- Original creations of TMM department scientists and postgraduates
- Original creations of Russian and Soviet engineers and scientists
- Models reconstructed by our foremen according to foreign scientists projects
- Copies of interesting models

Naturally, in telling about the Russian models of a collection that date back 150 years, it would make sense to give the history of Bauman University and the TMM Department (earlier Applied Mechanics Department), mention some reasons for the creation of the collection, contents of TMM course, scientific and pedagogic schools and their founders. The history of the discipline is at the same time the history of the scientific and pedagogical schools of the TMM department which had a serious influence on bath Russian and Soviet TMM scientific schools. It is enough to say that the scientist I. Artobolevsky who was well known in the scientific world community was the only Soviet scientist to be awarded J. Watt's medal, was a student of Professor Mertsalov and attended his lectures in Bauman University over 20th years of the 20th century as a private person.

These reasons and long discussions of authors resulted in the following. It is impossible to manage without the history of the university. However, it should be brief. It should be reduced to chronology and comments on it. The history of the formation of the discipline is connected both with the history of the development of the university, and

with the influence of various scientific schools. The first stages of the formation of a course were influenced by French and German pedagogical schools. However, development of applied mechanics was influenced by the traditions and features of the university's development, and features of Russian life.

During the preparation of this book we were kindly supported by the rector of Bauman University Professor Igor Fedorov.

We thank the employees of the University Museum and its director Galina Bazanchuk for presenting materials on history of the university, Ass. Professor Ilya Volchkevich and chief of Public Relations Department Alexander Kochanov for supplying photographic materials on the history of university.

In the book, photos of models executed by students Sergey Glushkov, Aleksey Ruzavin, Dmitry Plehov and the authors of the book are used.

We thank engineer Zynaida Tarabarina, students Michael Golenkov, Alexey Zinyagin, Julia Kozhuhova, Airat Rashitov and Tamara Tatyan for helping in making ready the book .

The Preface, Introduction, Chapters 1.1–1.4, 2.3, 2.4 were written by Professor A. Golovin and Chapters 1.5, 2.2, 3.1, 3.2.2–3.2.4, 3.3–3.8, 3.10, 3.11 were written by Ass. Professor V. Tarabarin. Other sections of the book were written by both authors.

INTRODUCTION

1. Brief history of Bauman University (IMTS–BMSTU)

Bauman Moscow State Technical University is one of the oldest Russian universities. By the RF President's Decree issued on January 1995, Bauman Moscow State Technical University was recognized as a Greatly Valuable Object of the Russian Federation Peoples' Cultural Heritage. In 2005, BMSTU celebrated its 175th anniversary.
July 1, 1830 is considered to be the official date of the university's establishment. That day, Emperor Nicholas I signed the project of "Edict about Moscow Educational Industrial Institution" statement. By this edict, the Founding Hospital acquired the status of a Technical University. However, its history dates back to the middle of the 18th century.
The history of the university is described rather circumstantially in a number of books, from which [1–4] are supposed to contain most detailed description. Therefore, we decided to limit the introduction to a brief chronology, singling out the dates and events that concern the TMM department, TMM discipline, the collection of mechanisms.

1.1. Prehistory: Founding Hospital (1763–1830)

June 10, 1763 Russian Empress Catherine II was introduced to "The general plan of the Imperial Founding Hospital". Moscow Founding Hospital was intended for the orphans Russian subjects. Its founders had the idea that the Hospital was to provide technical education sufficient for running factories and plants for its charges. On April 21, 1764 Catherine II laid the first stone of the Founding Hospital. That very day the enrolment of foster children began, the first two of them being named Catherine and Paul after the Empress and her son.

Fig. 1: The Slobodskoy palace in XVIII century [5]

A. Golovin and V. Tarabarin, *Russian Models from the Mechanisms Collection of Bauman University.*

1792 – In the Industrial workshops of the Founding Hospital, 257 children were studying the trades of carpenter, bench work, blacksmith work, tin, copper, aurum and silver, turner, watchmaker, engraving, toolmaker, carriagemaker, saddlemaker, shoemaker, typography, bookbinding, baker, glovemaking, haberdashery. Altogether 20 professions were taught.

MSTU became an independent educational institution on October 5, 1826 when the dowager empress Mariya Feodorovna issued an Order to establish "large workrooms for different trades with bedrooms, dining-hall and other needs" the attached to the Moscow Founding Hospital in Nemetskaya Sloboda (Foreign Settlement). Emperor Nicholas I transferred the ruins of Slobodskoy Palace (Fig. 1), which was burned out during Napoleon's invasion in 1812 when the Moscow Fire occurred, to the institution then being organized.

1829–477 foster children studied trades in the Founding Hospital.

Fig. 2: Street Koroviy Brod ("Cow ford" in English) and Moscow Educational Industrial Institution [5]

1.2. Moscow Educational Industrial School (1830–1868)

1830, July 1 – the "Edict about Moscow Educational Industrial School" was issued (Fig. 2). By this edict the Founding Hospital acquired Technical University status. The date is considered to be the official date of the university's establishment. For conducting the studies, a teaching staff of 4 professors, 10 teachers, 1 mechanic, 18 craftspeople and 1 model, and a physics laboratory and library supervisor was established. Six-year education was provided. There were two educational courses available: mechanical and chemical.

In 1832, a physical laboratory, model cabinet, library and chemical laboratory for printed fabric pattern designing began their work. For practical training, mechanic, forging, joiner's and models', turner's, painting, pasteboard, tin- and copper casting, tin-smithery, engraving and bronze workshops were organized. Gradually, as the status of the institution was changing, the system and standards of education were changing,

too. In 1832, in addition to the primary course (reading and writing, arithmetic, calligraphy and choir singing) the teaching of geometry and initial specifications of algebra began.

In 1834 higher courses, meant for training not just craftspeople but erudite foremen, were begun. So, physics, chemistry and mechanics courses were included in the curricula.

In 1838, a mechanical workshop was opened and the teaching of practical mechanics and chemistry begun and two years later descriptive geometry was included. In 1839 the first graduation of erudite foremen took place.

In 1844, cast-iron and model workshops were opened. Theoretical education included the following subjects: religion, calligraphy, Russian grammar, German, arithmetic, geometry, algebra (up to higher degree equations), planar geometry, descriptive geometry applied to perspective view drawing, practical mechanics, physics and chemistry applied to crafts, brief Russian geography course, drawing and inking.

There were three students' grades: preparatory, workers' and foreman. On the completion of an integrate a theoretical and practical training course foreman, grade students graduated being given the rank of factory and plant running foremen. A small plant attached to the institution, which was equipped with the most modern tools for those times and machines, was established. Besides serving educational purposes, this fact permitted it to fulfil orders of industry.

1854 – Professor A. Yershov created the first Russian manual on applied (practical) mechanics "Foundation of kinematics, or elementary theory of motion in general and of mechanisms of machines in particular"[6].

1857 – expansion of the practical mechanics course took place.

In 1859–1867 A. Yershov was Head of the School. It owed to him not only the expansion of the theoretical education course but also bringing it up to university standard. From the end of the 1850s, all theoretical disciplines could be taught by advanced degree holders only and courses were at least to correspond with those of the university. Henceforward, the majority of theoretical discipline teachers (of mathematics, mechanics, physics, chemistry) were Moscow University graduates.

Fig. 3: The breastplate of graduate's the IMTS–MHTS–BMSTU [5]

1.3. Imperial Moscow Technical Secondary School (1868–1917)
July 1, 1868, Imperial Moscow Technical Secondary School was affirmed by law to be a Supreme Educational School). It had the following divisions: mechanical, chemical, mechanical-building. Some 310 students attended the IMTS. Henceforward, the graduates were given the rank of civil engineer, mechanical engineer, or processing engineer.
1870 – the IMTS students' uniform and the IMTS graduate's badge were approved, the IMTS participated in the first all-Russian manufacturing show in St. Petersburg and was awarded first prize (Fig. 3).
1870 – Professor N. Joukovsy began to teach in the IMTS.
1872 – the IMTS participated in the international show in Vienna, four prizes won.
1872–1892 – Professor F. Orlov taught "Application mechanics" course (Part 1 – Theory of mechanisms; Part 2 – Theory of machines) in the IMTS.
1873 – the first lithographic edition of lectures "Application mechanics" [7] was published by Professor F. Orlov.
1876 – the IMTS participated in the World Fair held in Philadelphia The system of practical training of engineers was the Fair's exhibit. The IMTS teaching methods were awarded a medal. The methods were based on three constituents, which were given rise to by all the prehistory of the institution, and which still remain today. They are:

- Practical training based on students' real work under conditions which are as close as possible to their future practice as is possible.
- Theoretical discipline studies being up to classic universities standards.
- Mutually beneficial relations of the higher technical school with industry.

1877 – the Charter of the first Russian Polytechnic Community established on the IMTS graduates' initiative was approved.
1878 – the IMTS participated in the international show in Paris, two highest awards and a gold medal won.
1892–1898 – Professor D. Zernov taught a "Application mechanics" course (Part 1 – Theory of mechanisms; Part 2 – Theory of machines) in the IMTS.

Fig. 4: Building of mechanical institute IMTS (founded in 1904) [5]

1895 – the lithographic edition of lectures “Application mechanics” was published by Professor D. Zernov [8].
1900 – the IMTS participated in the international exhibition in Paris and was awarded the Grand Prix. 1904 - it was build up the building of Mechanical Institute IMTS (Fig. 4)
To understand further the destiny of the University we should give the insight of political situation in Russia at the beginning of the 20th century and several dates which define the situation:

- Defeat in the Russo-Japanese war in 1904.
- Shooting down of the orderly demonstration in St. Petersburg, January 9, 1905 (Bloody Sunday – in Russian: Krovavoye Voskresen’ye).
- Tsar’s manifesto of October 17, 1905, which granted some freedom, and its discredit.
- The first Russian revolution of 1905 – December rebellion in Moscow and defeat of the revolution.
- On January 12, three days after Bloody Sunday the IMTS students’ meeting adopted the resolution that urged active revolutionary work. January 13 students decided to stop their studies. The studies were actually resumed on April 17, 1906.

August 27, 1905 the institutes of higher education reform decree was issued. The decree initiated their academic freedom and self-government. The posts of rectors and deans became elective. The first IMTS rector elected was A. Gavrilenko. When he was 12, his father sent him to the IMTS preparatory courses. After graduating from the IMTS, he was sent to N-AUS to perfect his skills and knowledge in the field of mechanical engineering. From 1888 until his death in 1914, A. Gavrilenko worked in the IMTS.
Contemporary name of the IMTS “Bauman Moscow State Technical University” is concerned with the year of 1905. Nicholai Bauman, who was a revolutionist, was killed on October 18, 1905 when the IMTS yard at the head of a demonstration leaving, which was to free prisoners of conscience from Taganskaya prison. 1914 – there were two divisions in the IMTS (or departments since 1917), 2,700 students, 93 teachers, 164 personnel. 1914–1918 – First World War.

1.4. Moscow Higher Technical School (1917–1930)

February, 1917 – demise of Emperor Nicholas II
March 6, 1917 – the IMTS was renamed the Moscow Higher Technical School (MHTS)
October 1917 – Great October Russian Revolution (in Russian – Velikaya Okty-abr’skaya Sotsialisticheskaya Revolutsiya)
The events of the year of 1917 made it possible to get higher education for great masses of population – for working class and the peasantry. However, the majority of population were completely illiterate or semi-literate. Not accidentally, in succeeding years the so-called “likbezy” – special organisations, which conducted campaign against illiteracy, were established. To study at a high school, a sufficiently higher level of grounding was needed. So, in 1919 special preliminary courses called “working faculty” (in Russian – “Rabochy Fakul’tet” – “RabFak”) were established. Many famous Soviet engineers and scientists studied there. Certainly, it was a shift and RabFak didn’t provide the fundamental grounding that was typical of the pre-revolutionary period. A decline in the general standard of education caused reconsideration of the curriculum and methods of technical problem solving became more specific.

1918 – electrotechnical and civil engineering departments were established, tuition fees were abolished.
Soviet power considered the creation of new technical intellectuals to be one of the most important aims. We can see the seriousness of the problem by the appointment of Nicholai Gorbunov, who was Sovnarkom's (ministry) business-administrator and Lenin's private secretary, as principal of MHTS.
1923 – there were 319 professors and teachers, 3,500 students, 40 laboratories, 7 workshops, 15 rooms, 3 museums, and 5 libraries in MHTS. Student composition by percentage: workers – 36%, the peasantry – 7%, white-collar workers – 53%, others – 4%.

1.5. Bauman Moscow Mechanical Machine-Building Institute (1930–1943)
1930 – MHTS was divided into five independent institutes of higher education:
- Moscow Mechanical and Machine-Building Institute (MMMI), which was named after N.E. Bauman.
- Higher Aeromechanic School, later Moscow Aviation Institute.
- Higher Civil Engineering School, later Moscow Civil Engineering Institute.
- Higher Chemical Engineering School, later Chemical Engineering Military Academy.
- Higher Power Engineering School, later Moscow Power Engineering Institute.

In 1932, cold and hot working, heat and flow machines, general machine-building, technical and economic departments were established in MMMI. In 1933, welding engineering department and, in 1936, instrument engineering and motor-vehicle and tractor departments were established.
June 22, 1941 – Germany commits aggression against the Soviet Union.
October, 1941 the majority of MMMI staff (1,100 students, 120 professors and teachers) were evacuated to Izhevsk (near the Urals). Two chairs of shell and cartridge manufacture (technology chair and equipment chair) were organized there. The professors worked as consultants at defence industry plants. 515 students and 45 professors remained in Moscow. More than 1.5 million ammunition units, about 60,000 of shooting equipment barrels, about 8,000 field mortar assembly units, about 300 antitank guns and a large number of other optional battle-front needs production was made at the MMMI manufacturing workshops.
April 1943 – re-evacuation from Izhevsk.
May 1943 – the name Bauman Moscow Higher Technical School was revived.

1.6. Bauman Moscow Higher Technical School (1943–1989)
1945 – MHTS departments were: mechanical engineering (565 students), heat and flow machines (526), fine mechanics (465), artillery (297), tank (289), ammunition (130). Later those departments were reorganized into mechanical engineering, heat and flow machines, instrument engineering, design, machine-building, and mechanical departments.
In 1945, MHTS became the first institute of higher education where a student Scientific and Technical Society was organised. In 1950 it was named after the prominent Russian mathematician and mechanic N. Joukovsky, who was a Bauman University' professor in 1870@1921. In 1948@1954 the number of MHTS students increased from 3,900 to 9,300. A number of new departments, which were meeting the demands of the USSR national economy (plasma energy installations, aerospace systems, rocket, electronic computer technologies and other departments), were established.

1.7. Bauman Moscow State Technical University (from 1989 up to the present)

In 1989, Bauman MHTS was reorganized into Bauman Moscow State Technical University (BMSTU) (Fig. 5).

Fig. 5: New training laboratory building BMSTU

In 1991, the most difficult stage of Russia's contemporary life began. Living standards of the Russian people fell, a brain drain began and entire branches of industry were lost. Under the circumstances, the MSTU rector managed not only to "beat the University about" but also to complete a new University building.

January 24, 1995, by Presidential Decree, Bauman Moscow State Technical University was been included in the State List of Valuable Objects of the Russian Federation Peoples' Cultural Heritage.

Nowadays there are the following departments in the University:

Scientific and training centres:

- Fundamental science
- Special mechanical engineering
- Robotics and complex automation
- Information theory and control systems
- Mechanical engineering technologies
- Radio-electronics, laser devices and biomedical technologies
- Power engineering

Departments:

- Social science and humanities
- Engineering business and management
- Military education
- Linguistics

- Sports and sanitary department
- Inter-branch institute of advanced training and retraining in new engineering and technology lines of development
- Branches of the university located in Dmitrov and Kaluga

Many outstanding Russian scientists and engineers taught or actively cooperated at the university.

The great Russian mechanic, N. Joukovsky, on graduating from Moscow University, since 1872 had been working as teacher of mathematics. In 1878 he was based the Department of Theoretical Mechanics. His works in the field of mathematics, mechanics and aircraft are known to scientists of the whole world. Among his pupils, there were many outstanding scientists and engineers.

Well-known mathematician and mechanic S. Tchaplygin, who had been taught in the IMTS in the Department of Theoretical Mechanics from 1896 till 1906, was a pupil of N. Joukovsky. Professors N. Mertsalov and L. Smirnov, educated in IMTS – MHTS original schools of applied mechanics, were disciples of N. Joukovsky. V. Goryachkin, a graduate of IMTS and Moscow University, the founder of the science of agricultural machinery and academies of agricultural mechanical engineering was his disciple.

In 1876, a disciple of N. Joukovsky, V. Shuhov – one of the greatest engineers in the world graduated from IMTS. His spatial designs, including Shuhov's hyperboloid tower, are known. He invented a cracking-process, created methods of calculation of cracking-devices, oil storage, bulk-oil barges, gas-holders and other devices, and constructions for the extraction, processing and transportation of oil and its products, an atomizer for burning liquid fuel (from oil up to black oil). He was the pioneer of oil hydraulics, the theory of direct-action and inertial pumps. In the field of economy, he offered simultaneously and irrespective from Lord Kelvin a coefficient which now can be called the "normative coefficient of economic efficiency of capital investments". The future Nobel winner and president of the USSR Academy of Science, S. Vavilov, was been taught physics from 1920 till 1923 in the MHTS.

Graduates of MSTU were the great Soviet engineers S. Korolev – the main designer of space-rocket systems, N. Pilyugin – the founder of gyroscopic systems of flying devices, A. Tupolev – an outstanding Soviet aircraft designer.

Graduates of the University are nine cosmonauts-astronauts (data on 2005): Konstantin Feoktistov (12th) astronaut of the world), Alexey Eliseev (37th), Oleg Makarov (65th), Gennady Strekalov (99th), Alexander Alexandrov (123rd), Vladimir Solov'ev (136th), Alexander Laveykin (200th), Alexander Balandin (226th), and Elena Kondakova (317th).

It is necessary to mention separately the cooperation of the great Russian mathematician P. Chebushev with IMTS. His cooperation has began with a response to the program of a course of practical mechanics of A. Yershov. In the collections "Reports and the speeches said at solemn assembly of Imperial Moscow Technical School" are published his articles "About the centrifugal equalizer" (1871) and "About cog-wheels" (1872). P. Chebushev cooperated with F. Orlov. Models of mechanisms of P. Chebushev are stored in the MSTU museum and in the collection of mechanisms, presented by him and made in workshops. After the recognition of his merits before IMTS, P. Chebushev was selected by the Honorary Pedagogic Councilor of IMTS, Honorary Member School's Pedagogic Council.

1. "APPLIED MECHANICS" – TMM IN BAUMAN UNIVERSITY

The history of the forming of the discipline is linked with both the university's development history and the influence of various scientific schools. The first stages of the course-forming were influenced by French and German pedagogic schools. But traditions and special features of the university's development and peculiarities of the Russian way of life also affected the development of applied mechanics course. From this aspect, mechanical workshops being available (the fact nowadays called the material resources availability) had their influence on the forming of the "Russian system" of applied mechanics. As the university was developing, it was promoting and student and teaching staff were recasting. All these facts affected the contents of the applied mechanics course. Analyzing the contents of the course over the lifetime of the university, we can distinguish four stages of forming of the applied mechanics – theory of mechanisms course.

Stage 1. First stage dates back to 1838. That year teaching of the Practical Mechanics course began. In 1857, through Professor Alexander Yershov's efforts, expansion of the Practical Mechanics course took place. We can assume that Russia's defeat in the Crimean War (1853–1854) contributed to this fact. It was perhaps the first war of the 19th century won, not by military leader skills and soldier heroism but by material technical superiority of one of the combatants. We can see the course contents from the first Russian manual published in 1854 by Professor Yershov for Moscow University and Bauman University students [6]. The course was merely descriptive. Its theoretical part included the most elementary knowledge in theoretical mechanics and its contents corresponded to students' grounding.

Stage 2. Since the end of the 1850s Professor Yershov's efforts (he had been the principal of the university since 1859), all theoretical subjects could be taught by Master's degree holders and the course contents were to correspond with those of the university. It should be mentioned that until quite recently before the development of computer engineering TMM as a science and a discipline was mainly the apparatus for the analysis of mechanisms. Synthesis problems were relegated to the background because of the laboriousness of computations. It was natural to use graphical methods. However, students were educated well enough to understand analysis methods. The stage was linked with the names of Professors F. Orlov, D. Zernov, and N. Mertsalov. Two parts of lithographic courses of applied mechanics lectures (Part 1 – Theory of mechanisms, Part 2 – Theory of machines) by F. Orlov (1873–1892) [7], D. Zernov (1895) [8] and "Kinematics of mechanisms" and "Dynamics of mechanisms" by N. Mertsalov [14, 15].

Stage 3. The Great October Russian Revolution (in Russian – Velikaya Oktyabr'skaya Sotsialisticheskaya Revolutsiya, 1917) made it possible for wide circles of working class and peasantry to get higher education. Adults having workers' professions or peasant experience entered universities. Many of them went through the civil war. But their grounding was far worse than the grounding of those students of the pre-revolutionary period. So, special preliminary courses, called "working faculty" (in

A. Golovin and V. Tarabarin, *Russian Models from the Mechanisms Collection of Bauman University.*

Russian – “Rabochy Fakul’tet” – “RabFak”), were established to deepen their knowledge. In this connection, applied mechanics course contents was reduced to a certain number of problems. It was natural to use methods allowing students to bring the solving of a problem down to methods which require a minimum of calculations to be performed. Grapho-analytical methods, which make it possible to solve problems put using a number of graphical constructions and scale computations, were such kinds of methods. Professor L. Smirnov was an expert in these methods. Professors L. Reshe-tov, A. Savelova, V. Gavrilenko, and N. Skvortsova, future heads of TMM department, came of the pedagogical school which he established. In the course of time, workers and peasants were replaced by a students who had completed a school course of education. This fact made it possible to enlarge the course almost to the former, pre-revolutionary level. But graphical methods were still preferred to analytic ones and they proved to be more efficient until the 1960s and 1970s.

Aspirations to teach students how to solve TMM problems by one method or another were common features of stages 2 and 3. Naturally, there was no place for mechanisms multivariant synthesis problems in the course’s curriculum. Mechanisms properties were rejected. Besides, it was believed that there should be a common course for all the specialities of the university. As a result, a number of specialities rejected the course or agreed to have a reduced course because they believed (and often believed reasonably) that the course didn’t satisfy the needs of the speciality.

Stage 4 – the present. Since computer facilities appeared, it has become an especially serious problem for engineers and scientists to choose computation methods when designing a mechanism. For student computation problems have been relegated to the background. But mechanically, educational material and the course still pay much attention to computation procedures based on geometric methods. This approach is often unsuccessful, especially when solving sets of non-linear and differential equations. Mechanisms properties are still only touched upon. So, in prospect we can expect that the theory of mechanisms discipline will pay more attention to properties of mechanisms as a base for multivariant designing of mechanical devices of modern mechatronic systems. The second trend of the discipline’s development is to establish problem-oriented courses, which would content both the fundamental basis of TMM and satisfy the needs of the speciality.

1.1. The first course of the Theory of Mechanisms

Professor A. Yershov. Setting up of mechanisms collection

Fig. 1.1: Professor Alexander Yershov (1818–1867)

1.1.1. Scientific and pedagogic activities of Professor A. Yershov

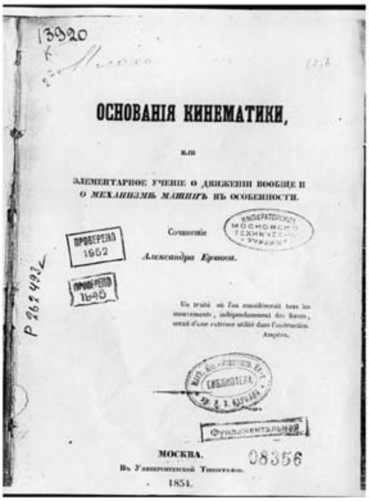
Alexander Yershov (1818–1867) (Fig. 1.1) graduated from the Mechanics and Mathematics department of Moscow University in 1839. He underwent his probation period in the engineering institutions of St. Petersburg, then deepened his knowledge and experience in Paris [7, 8]. In 1844 he defended his thesis “Water as engine”. That year he started teaching practical mechanics first as adjunct professor and then (from 1853) as an extraordinary professor.

Being the principal of Moscow Vocational School (he was professor of the School in 1845–1867, and its principal in 1859–1867), A. Yershov established laboratories there and proposed reorganizing the School into Moscow Highest Technical School. Soon after Yershov’s death, in 1868, Moscow Industrial School became Imperial Moscow Highest Technical School. In 1858, Yershov began delivering public lectures on some problems of practical mechanics in the School.

In 1854, A. Yershov published the first Russian systematic manual in applied (practical) mechanics, called the “Foundations of kinematics, or elementary theory of motion in general and of mechanisms of machines especially” (Fig. 1.2). Today this book could be called a manual in TMM.

In 1857 A. Yershov succeeded in pressing for the decision that IMTS theoretical disciplines could be taught by advanced degree holders only.

“Foundations of kinematics, or elementary theory of motion in general and of mechanisms of machines especially” [6] (266 pages, 227 illustrations). The fragment of the title is emphasizes of Yershov.



ОСНОВАНІЯ КИНЕМАТИКИ,

МОСКВА.

1854.

Fig. 1.2: Alexander Yershov, “Foundation of Kinematics...”

It ought to be mentioned that A. Yershov and later F. Orlov and D. Zernov, used the term “practical mechanics” term in their works regardless of the name of the course.

Monge’s mechanisms of the classification of machines elements based on the motion type transformation in terms of its mode and direction and Ampère’s concepts of the geometric and kinematic properties of mechanism being independent from forces were the foundations of the course. The influence of Ampère’s concepts was testified by the book’s epigraph:

Un Traitè ou I ‘on considèrerait tous les mouvements,
indèpendamment des forces,
serait d ’une extri me utiliti dans I ‘nstruction (Ampère).

Yershov’s “Practical Mechanics” course provided three natural stages for the study of the theory of machines:

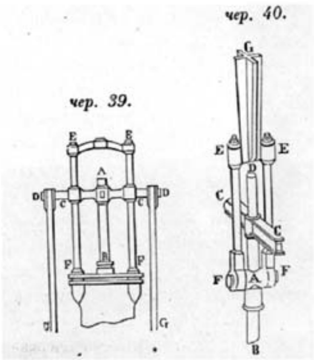
- Transmission and transformation of motion, or the mechanism of machines.
- Receiving and transmission of work of driving forces, theory of engines, dynamic theory of machines.
- Machine construction based on the strength of material’s theory.

In his “… mechanisms of machines…” manual [6], Yershov considers the first stage-transmission and transformation of motion, or machines mechanism. The manual consists of three parts: (1) elementary kinematics; (2) motion transmission systems; (3) motion transformation systems.

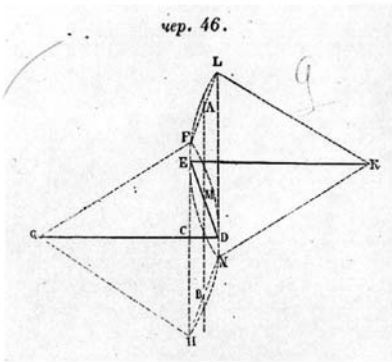
On the first pages (pp. 1–49) an elementary description of the foundations of mechanics and especially of particle and body kinematics was given. Also reference data considering machining speed limits of various materials was presented, which was to the point because the manual was intended for the use of Vocational School students. The substance of the manual was given in Part 2: “Transmission and transformation of motions. Theoretical ground of the material was primitively formulated. The part zeroed in on the description of the large number of different mechanisms. Historical information on pp. 50–54 contained Monge’s and Ampère’s concepts and general review of the manual.

Chapter 1. “Motion transmission” (pp. 54–179).

- Continuous rectilinear motion: pulleys and tackle blocks, hydrotransmission, wedge.
- Rectilinear wobbling motion (Fig. 1.3): guides, slider-conrod connection, pile-drivers, Watt and Evans’ straight-line mechanisms.
- Continuous circular movement (Fig. 1.4): endless ropes, belts, chains; transmission with parallel and non-parallel axes; variable diameter pulley; parallel link mechanism; involute and cycloidal gears; contact clearance; internal gearing, pin gearing; bevel gearings; skew gears; endless screw (worm gearing); Hooke’s joint; sliding joints (couplers and friction couplings); compound mechanisms (with fixed axles, planetary trains, clockworks).
- Circular wobbling motion: leverages and lever-and-gear mechanisms. Chapter 2. “Motion transformation” (pp. 180–256).
- Continuous rectilinear into continuous circular motion transformation and vice-versa transformation (Fig. 1.5a, b): simple and complex necks, screw and nut.
- Continuous circular into rectilinear wobbling motion transformation and vice-versa transformation (Fig. 1.5c–e): connecting-rod and crank, eccentric, Stephen-son’s double eccentric, Lahire’s planetary train, flat and spatial cam mechanisms, fric tiohammer, rack-and-pinion, compound gear (differential-rack gear).

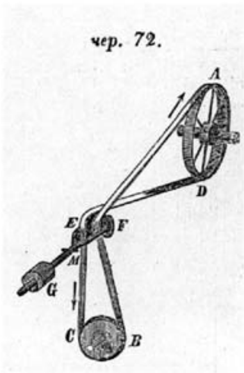


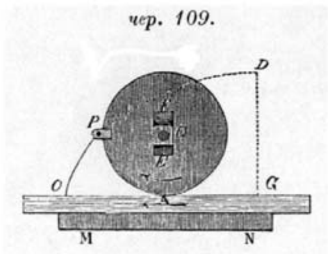
Guides of sliders



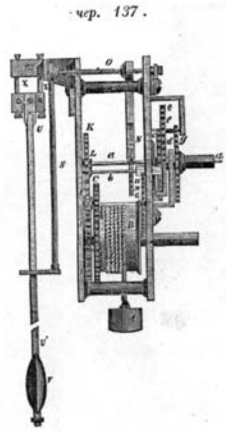
Watt's mechanism

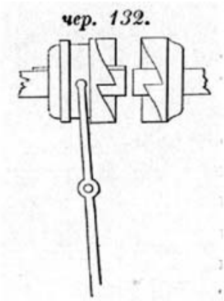
Fig. 1.3: Transmission of rectilinear moving



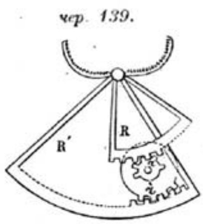


Drawing of involute





Couples



Mechanism for facing of piles

Fig. 1.4: Example of transmission of circular moving

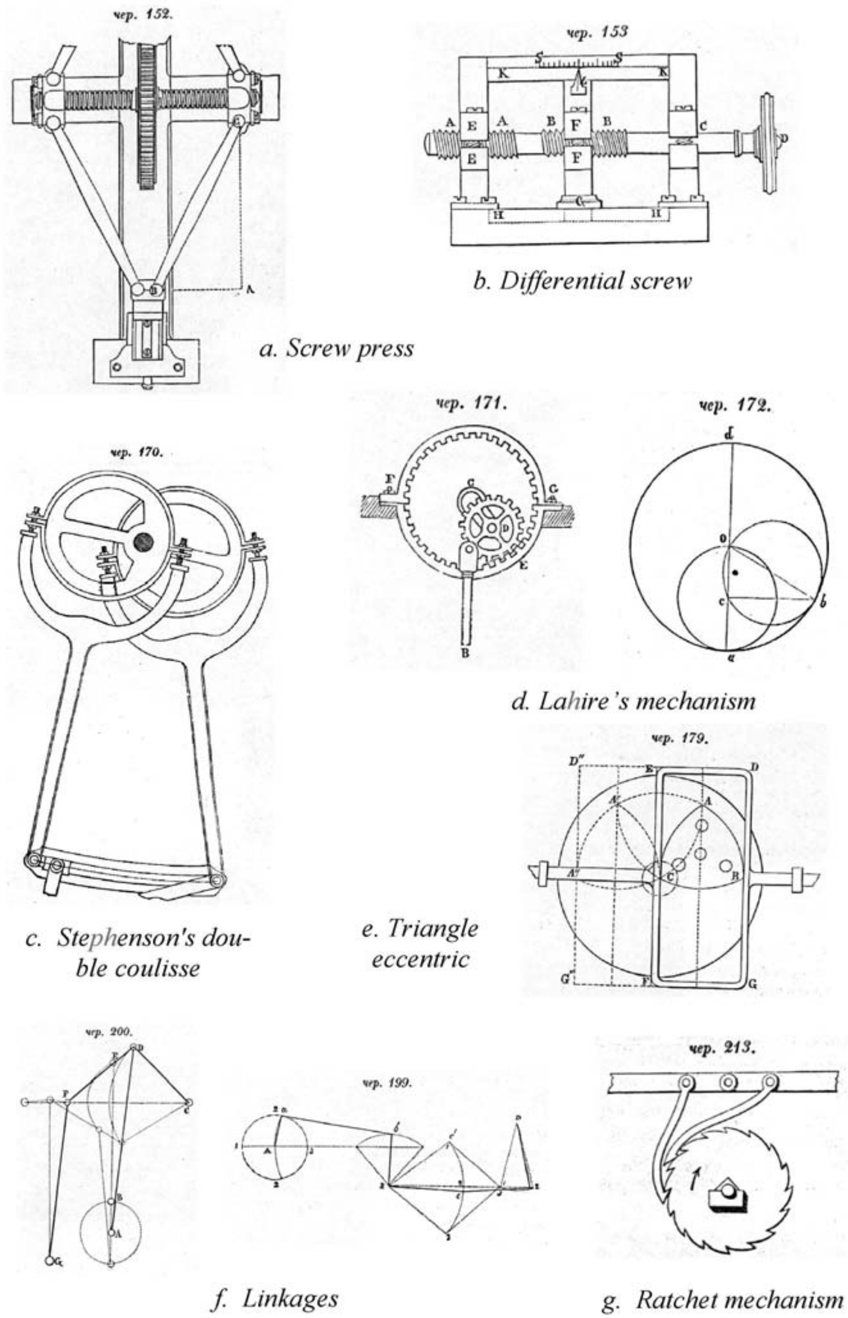


Fig. 1.5: Example of transmission of circular moving

• Rectilinear wobbling into circular wobbling motion transformation and vice-versa transformation (Fig. 1.5f, g): crank-and-rocker mechanism and oscillating crank gear, direct contact motion transmission (cam mechanisms for hammer block lift), ratchet-and-pawl mechanism.
• Continuous circular into circular wobbling motion transformation and vice-versa transformation.
Chapter 3. "Differential and compound motions" (pp. 257–266).

1.2. Enlargement and development of the course
Professors F. Orlov, N. Zernov, N. Mertzalov (1868–1929)

As has already been mentioned, from the end of the 1850s theoretical disciplines could be taught by advanced degree holders only and course contents were to correspond with those of the university. Since then the majority of academicians teaching theoretical subjects (mathematics, mechanics, physics, chemistry) were Moscow University graduates. Accordingly, a student's grounding was much in demand. It essentially enlarged. We can see from the fact that the level of "Theoretical mechanics" and "Analytical mechanics" [9, 10] courses, which N. Joukovsky began to teach in 1870, was very high. This fact permitted the development of a "Application Mechanics" course on a higher scientific level comparatively to the course of Yershov.
The "Application Mechanics" course was formed by Professors Orlov (1872–1892), Zernov (1892–1898), Mertzalov (1898–1929) one after the other. In this paragraph we briefly introduce the profiles of the professors, try to give an insight of the applied mechanics course and its evolution over a period of time.

1.2.1. Professor Feodor Orlov (1843–1892) [11, 12]

Fig. 1.6: Professor Feodor Orlov (1843–1892)

F. Orlov (Fig. 1.6) was a medical officer's son. He graduated from Moscow University and entered mathematical Department of Moscow University in 1859 and completed his studies in 1863. The same year was appointed to the pure mathematics chair. In 1869 he defended his Master's thesis "About the reciprocity of differential equations". That year

he was recommended to become head of Moscow University Practical Mechanics chair and was sent on an assignment to universities in Switzerland, Germany and France. Before this he had met with P. Chebyshev and J. Vyshnegradsky in St. Petersburg.
During 1869–1870 he attended lectures on Mathematics and Applied Mathematics in Switzerland (Zurich). There he translated Cheyshev’s paper about parallelogram for Civilingeneur magazine. The same year he was elected head of the IMTS Practical Mechanics and Thermodynamics chair. From the summer of 1870, Orlov attended the lectures of Professors F. Reuleaux and Cristofel at Gewerbe-Academie and Weyershtrasse and Cronecer at Berlin University. Then he removed in Belgium and attended lectures on Mathematics at the University of Liège till the end of 1871. Until the autumn of 1872 he was in Paris and attended lectures at the Central and Engineering Schools, Sorbonne, College de France.
In autumn 1872 he returned to Russia and from then till his death in 1892, he taught Practical Mechanics course at Moscow University and the IMTS. Joukovsky supposed that they were the best courses in Russia. In this period he began putting together the mechanisms collection of the universities.
The scientific interests of Orlov concerned pure mathematics (theory of differential equations, theory of surfaces, theory of rolls). His pedagogic activity was reflected in his course “Application Mechanics” [11]. He published the first lithographic bipartite course (Part 1 – Theory of mechanisms, Part 2 – Theory of machines) in 1873 and then republished the course more than once (Fig. 1.7).

1.2.1.1 “Applied Mechanics” of Professor F. Orlov [11]

Fig. 1.7: Feodor Orlov’s “Applied Mechanics” (Part 1)

Part 1 – Theory of Mechanisms (224 pages, 232 illustrations)

Part 1 consisted of an introduction, general theory of mechanisms and analysis of three classes of mechanisms classified according to connection type of their elements. Introduction. Classification of kinematic pairs according to their DOF. Lower and higher pairs. Mechanism is a system with DOF = 1. Like Professor A. Yershov, Orlov used two-level classification. The first level was classification based on the type of connection of mechanisms elements. The second level was the Willis classification based on a moving transformation type. Professor Orlov changed the sequence of the relation of materials and theoretical level as opposed to professor A. Yershov.

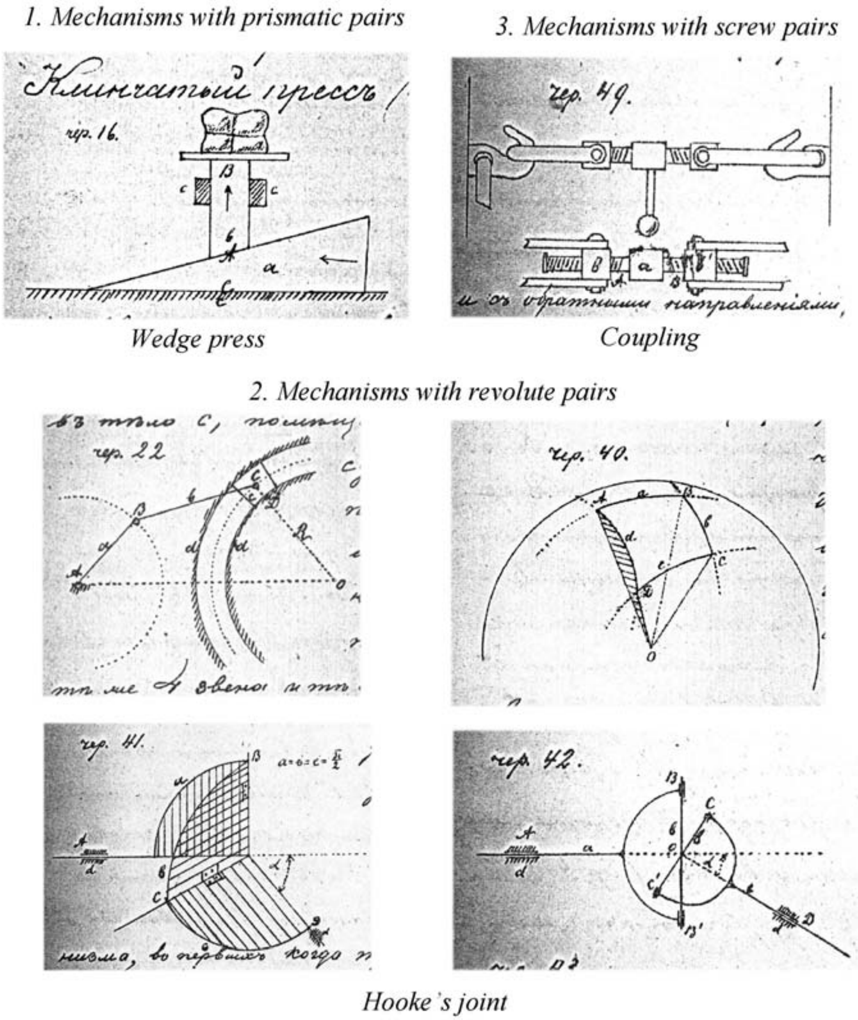


Fig. 1.8: Examples of mechanisms with lower kinematics pairs

General theory of mechanisms. Mechanisms with lower kinematic pairs. Mechanisms with prismatic kinematic pairs, which a very little. Wedge press, grip. Mechanisms with revolute pairs. Four bar linkage, parallelogram, antiparallelogram. Crank-slider mechanism. Ellipsograph, Oldhem’s clutch, sinus-mechanism, etc., Hooke’s joint and his analogues, control mechanism for railway semaphore. Mechanisms with screw pairs. Mechanism with three screw pairs. Practical application. Transformation of crank-slider mechanism. Examples of the mechanisms listed here can be seen in Fig. 1.8.
Three further classes of mechanisms classified according to the connection type of mechanisms elements (Willis’s classification) are considered.

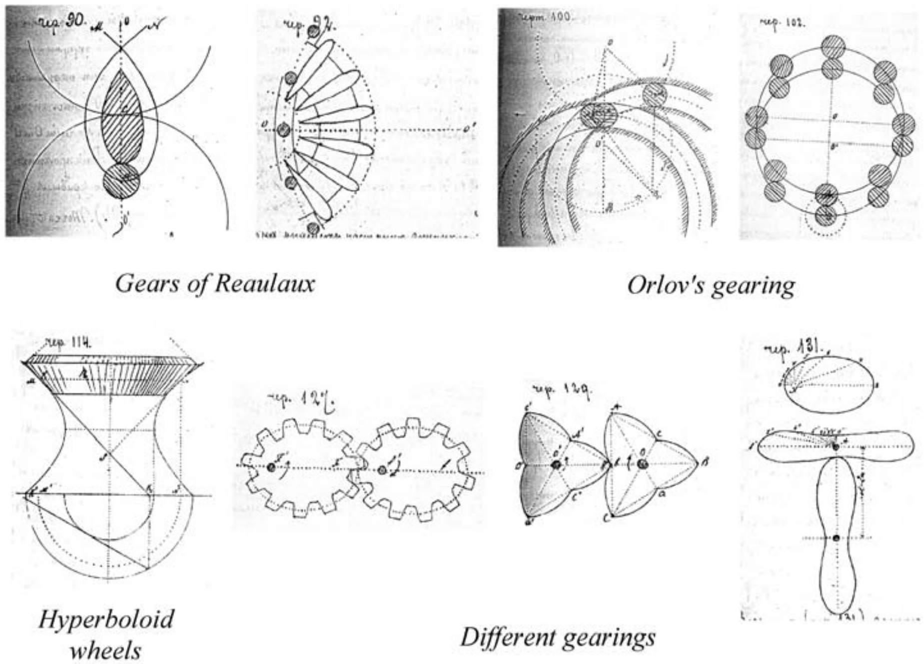


Fig. 1.9: Examples of moving transmission by direct contact (gearing)

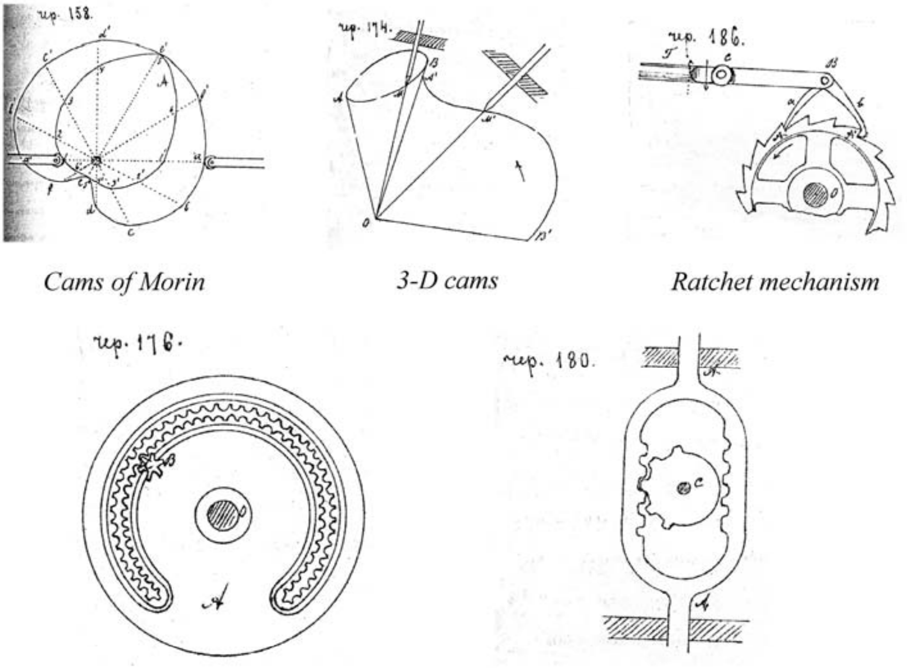
Class 1. Moving transmission by direct contact

- Willis’s momentary gear ratio theorem.
- Teeth-wheels (constant and variable gear ratio with constant indicium). The methods for profile tracing (Poncelet, Camus, Reuleaux). External and internal involute gearing. Cycloidal gearing. Pin gearing. Teeth profiling by two points. Teeth profiling by arcs of circle (Willis’s method). Hooke’s wheels. Rotation axes meet, bevel gearing. Skew rotation axes, hyperboloid wheels, screw and worm gearings. Unicycle elliptical wheels.
- Cam mechanisms (variable gear ratio with variable indicium). Examples of cams

with rectilinear and rotary moving. Types of cam mechanisms. Example of practical application.

- Mechanisms with constant gear ratio with variable indicium. Examples. Ratchet mechanisms, including clockwork.

Examples of the mechanisms listed here can be seen in Figs. 1.9, 1.10. A transmission designed by Orlov and manufactured at the IMTS workrooms is one of them. Unfortunately, it was impossible to find this model. It should be mentioned that Orlov gathered a considerable collection of mechanisms made in Germany and France.



Cams of Morin *3-D cams* *Ratchet mechanism*

Other mechanisms

Fig. 1.10: Examples of moving transmission by direct contact (cams and other mechanisms)

Class 2. Moving transmission by coupler

- Mechanisms with parallel axes. Four bar linkage (instantaneous center of zero-velocity, possible variants of moving transformation, parallelogram, antiparallelogram, Grashof s theorem). Crank-slider mechanism, sine-mechanism, ellipsograph. Crank -shaft and eccentric.
- Hooke’s joint.

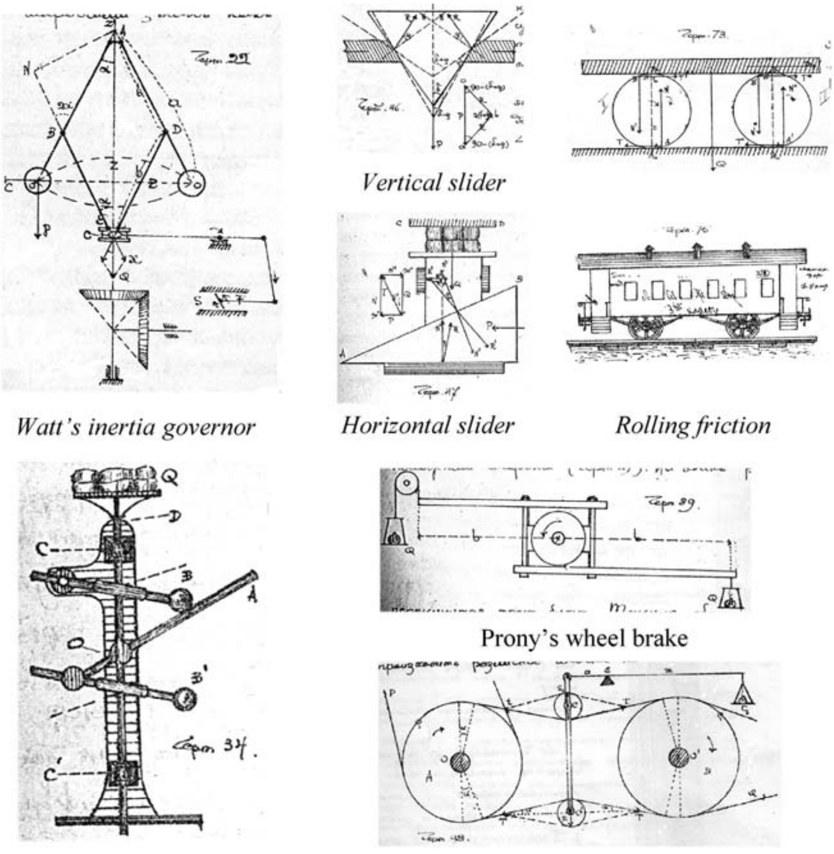
Class 3. Moving transmission by flexible elements or liquid medium

Belts and ropes.

Part 2 – Theory of Machines (344 pages, 247 illustrations)

This part accords with the second stage of “Applied mechanics” course [11], namely, the problems of friction and dynamic theory of machines (Fig. 1.11), theory of motors (Figs. 1.12, 1.13).

Dynamic theory of machines. Machine and mechanism. The first foundation of machine dynamics. Kinetic energy and dissipation. Speed control of machine and reason for using fly-wheel: gain in mass of fly-wheel or gain in mass of coupler, for example? About regulation: Watt’s inertia governor, parabolic and pseudoparabolic governor, etc. Problems of friction. General problems. Some devices that use the friction effect. Determination of friction coefficient experiment. Prismatic pair: wedge, horizontal and vertical slider. Revolute pair. Screw pair. Teeth-wheels. Transmission of work by friction: friction-wheels, friction couple, etc. Rolling friction and its application. Friction of flexible bodies. Transmission of work by endless belt. Brakes. Dynamometer of Navier, Imre, Hachette, Watt, etc.



Vertical slider

Watt’s inertia governor *Horizontal slider* *Rolling friction*

Prony’s wheel brake

Lifting jack The belt dynamometer of Briggs

Fig. 1.11: Some examples from Theory of Machines

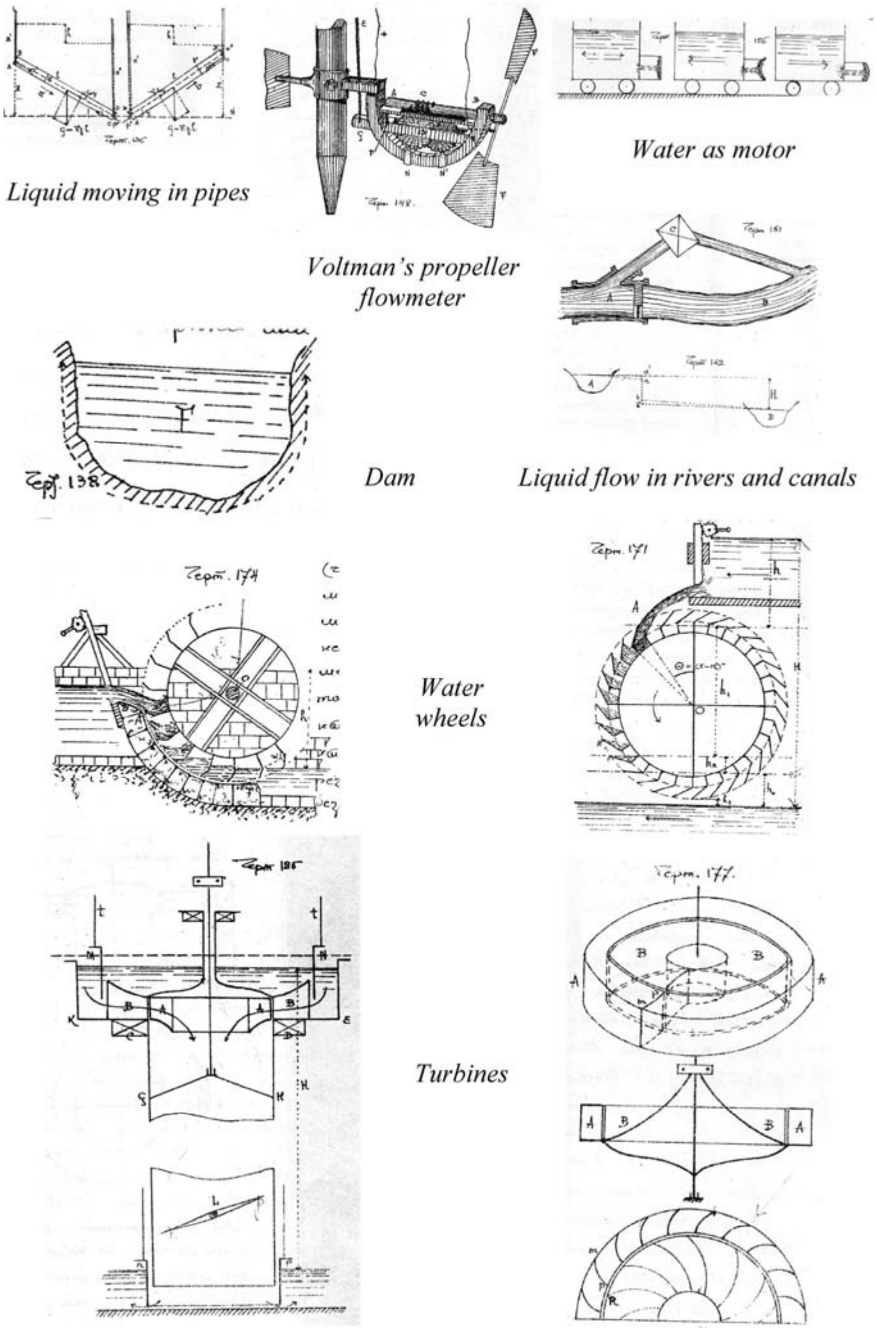
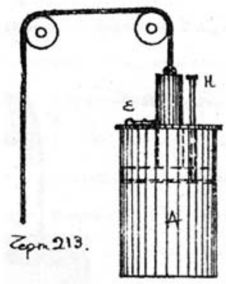
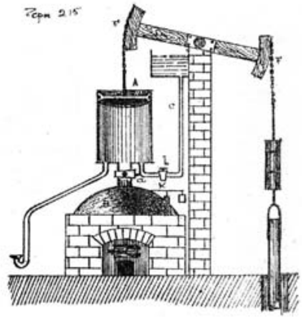


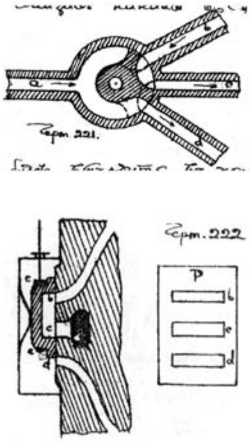
Fig. 1.12: Some examples from the Theory of Machines (motors) Hydraulics



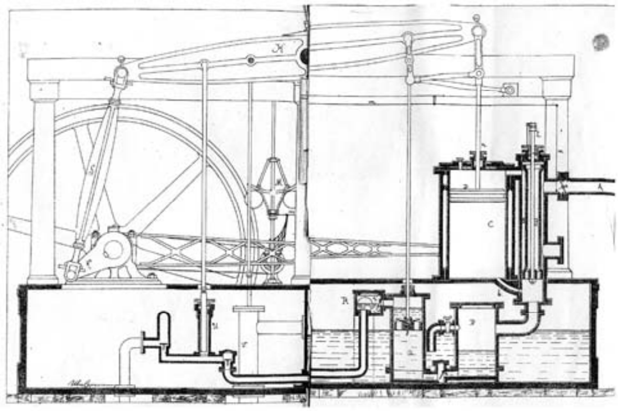
Steam engine of Papin (1690)



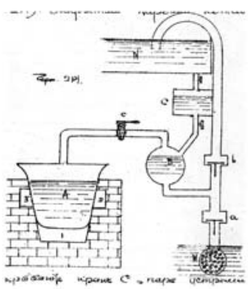
Steam engine of Newcomen (1705)



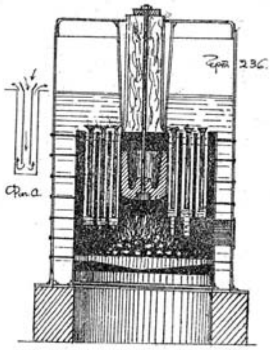
Steam box systems



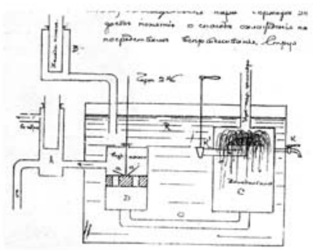
Steam engine of Watt



Steam engine of Savaery (1698). This type of machine installed in St. Petersburg in 1718



Boiler of Field



Refrigerator

Fig. 1.13: Some examples from the Theory of Machines (motors)

Examples illustrating the problems of friction and dynamic theory of machines can be seen in Fig. 1.11.

Theory of motors

- *Living motors*
- *Hydraulics*

Hydrostatics. Mains pressure. Equilibrium of body submerged in liquid. Equilibrium of floating body. Determination of mains pressure ponderable liquid.

Hydrodynamics. Flowing pressure. Liquid streaming from filler. Liquid moving in pipes. Liquid flow in rivers and canals.

Water as motor, dams.

Hydraulics machines. Water wheels. Turbines (eight types).

Air as motor.

Examples considering this part of the course are shown in Fig. 1.12.

- *Thermodynamics*

General questions. About steams. Caloric machines. Work measuring of steam engine.

Steam engines. Papin – Savary steam engines. Installation of Papin's steam engine in Peter the Great's garden in Petersburg in 1718. Newcomen and Kawlay. Watt's steam engine.

Systems of steam engines and devices. Steam boxes. Furnace, steam boilers. Outfits of boilers (accessories). Manometers, float-gauges, float level gages, safety valves, feed of boiler.

Refrigerators.

Examples considering this part of the course are shown in Fig. 1.13.

1.2.2. Professor Dmitry Zernov (1860–1922) [13, 14]

D. Zernov graduated from Moscow University and Petersburg Technological Institute.

1892–1898: professor of the chair of Application mechanics and Thermodynamics of the IMTS (Fig. 1.14).

After 1898 he was a rector of Kharkov and St. Petersburg Technology Institutes.

Fig. 1.14: Professor Dmitry Zernov (1860–1922)

1892–1898 – Professor Zernov taught Application Mechanics courses (Part 1 – Theory of Mechanisms; Part 2 – Theory of Machines) in the IMTU.
1895 – The lithographic edition of Application Mechanics lectures by Professor Zernov. The course was enlarged and republished more than once. The last edition of the “Theory of Mechanisms and Machines” manual came out in the 1930s.

“Applied Mechanics”, Part 1 – Theory of Mechanisms (432 pages, illustrations); Part 2 – Theory of Machines (232 pages, illustrations) [13].
Part 1. Theory of Mechanisms. The attempt to compare the presentation of the Theory of Mechanisms in Orlov and Zernov’s manuals was unsuccessful. The volumes and configuration of their contents were different. Conceptions of the course were different, too. We can mention the fact that the course was enlarged, mathematical and the theory of mechanics apparatus was complicated. Specifically, methods of radius of curvature determination were included in the course.
Presentation began with planar moving. Trajectory. Velocity and acceleration of points. Center of curvature of rolls. Savaery’s theorem (two methods for center of curvature determination).
Then the questions of the following type were considered. Kinematics pairs and their elements. Lower kinematics pairs. Kinematics chain and mechanism. Force closing of pairs.

Отдѣлы.	1 классъ. a). Отношеніе скоростей *постоянно.*	2 классъ. a). Отношеніе скоростей *перемѣнно.*	3 классъ. a). Отношеніе скоростей *постоянно* или *перемѣнно.*
	b). Направленія движеній ведущаго и ведомаго звеньевъ *постоянны* или мѣняются *одновременно.*		b). Направленіе движенія одного звена *постоянно*,—другого мѣняется *періодически.*
I. Передача движенія *непосредствснным прикосновеніемъ.*	*Фрикціонныя колеса.* *Зубчатыя колеса.*	*Некруглыя зубчатыя колеса.*	*Эксцентрики и кулаки.*
II. Передача движенія *промежуточными твердыми звеньями.*	*Совокупность зубчатыхъ колесъ.* *Параллельный кривошипъ.*	*Антипараллельный кривошипъ.* *Шарниръ Гука.*	*Кривошипъ и шатунъ.* *Направляющіе механизмы.*
III. Передача движенія *гибкими или жидкими тѣлами.*	*Шкивы и барабаны для ременной, канатной и т. п. передачъ.*	*Коноидальные барабаны.*	

Fig. 1.15: Zernov’s classification of mechanisms

Planar linkages. Planar four bar linkage: Grashof s theorem, dead points of mechanism, Four-bar linkage, paralellogram with revolute pairs. Four bar chain transformation. Mechanism of steam engine ordinaries and wobbly. Graphic and analytical methods for search of positions, velocities and accelerations. Eccentric. Rocker mechanism. Sinus-mechanism. Ellipsograph. Oldhem's couple. Wedge. Machine-tool of Leonardo da Vinchi. Chain with three screw pairs. Mechanisms which transformed from chain with three screw. Flexible and liquid bodies as links of kinematics chains.

A description of mechanisms was carried out according to Zernov's classification analogous to Willis's "Classification of mechanisms" based on the character of moving transformation (Fig. 1.15).

Here examples of theory presentation in accordance with Zernov's classification are given.

Mechanisms of class 1, group 1

- Cylindrical teeth-wheels. Texture of teeth-wheels. The methods of fitted profiles building. The method of enveloping curve determination.
- Method of enveloping poloid for teeth building. Cycloidal and involute gears. Method of equidistant curves for tooth profile building.
- Particular case of gearing. External and internal gearing (involute and cycloidal). Involute gearing. Pin gearing. Teeth profiling by arcs of a circle (Willis's method). Hooke's wheels. Bevel gears. Hyperboloid wheels, screw and worm gearings. Rack gearing. Teeth tracing on straight line for one wheel and on equidistant epicycloids for another. Combination with bidentate wheel (Rutt's ventilator).

Mechanisms of class 2, group 2

- Moving transmission by direct contact. Willis's theorem about momentary gear ratio.
- Moving transmission by solid (coupler).
- Moving transmission by flexible banding.
- Transmission of rotation by roll curve. Dynamics of machine.
- Elliptical wheels.

Mechanisms of class 3, group 1

- Cam mechanisms. General case. Cordiform eccentric. Eccentric of Morin. Eccentrics in frames (circular eccentric, eccentric in the form of an arc triangle).

Mechanisms of class 3, group 2

Elliptical directive mechanism: mechanism of Evanse, triangle Reuleaux. Conchoidal directive mechanisms. Chebyshev's directive mechanism. Lemniscate directive mechanism of Watt. Inverse directive mechanisms. Precision directive mechanisms. Directive mechanism of Gart, Peaucelier. Pantograph. Watt's parallelogram.

Mechanisms of class 3, group 3

Moving transmission by flexible bodies: moving transmission by endless cord with constant ratio; disposition of cords. Ratchet-wheels. Clockworks (anchor-handling gear, etc.).

Part 2. Theory of machines. The content and structure of this part substantially repeats the material of Orlov's book. Therefore, we mark the fragment which enlarges Orlov's lectures and proposes new advances in science.

Steam engine.

Dynamics of machine.

Steady motion of machine. Condition of balance of forces, acting in machine.

Compensation nonuniformity of machine's motion. Fly-wheel. Determination of tangential component of force. Work of tangential component of force. Work of resisting force. Determination of fly-wheel mass.
Regulators. Principle of operation. Conical pendulum. Insensibility and non-uniformity of regulator. Centrifugal governer of Watt and six types of regulators. Friction.

1.2.3. Professor Nikolay Mertsalov (1866–1948) [15]
Professor Mertsalov, a student of prominent Russian engineer N. Jukovsky, graduated from the Mathematics faculty of Moscow State University in 1888 (Fig. 1.16). He worked at one of the German plants and attended lectures at Dresden Supreme Technical School. Mertsalov returned to Russia in 1892. He passed his Master's degree examinations at the Imperial Moscow Technical School and became Mechanical Engineer in 1894. In 1895, at Joukovsky's invitation, Mertsalov became Associate Professor at the IMTS and supervisor at its Applied Mechanics laboratory. At the same time he taught Applied Mechanics at Peter's Agricultural Academy. After 1897 Mertsalov was an adjunct-professor of the Applied Mechanics Department. Besides his work on the creation of applied mechanics course, Mertsalov taught fourth-year students of the IMTS design of water turbines and carried out internal-combustion engine research experiments.

Fig. 1.16: Professor Nikolay Mertsalov (1866–1948)

After 1921, Mertsalov combined his work at the IMTS with that at the Applied Mechanics Department of Timiriazev (former Peter's) Agricultural Academy. It should be mentioned that in 1923 the famous soviet scientist Ivan Artobolevskiy attended Mertsalov's lectures at BMSTU as a private person. He left BMSTU in 1929. At the end of the 1930s, Mertsalov founded a Theoretical Mechanics seminar at the Institute of Machines Research of the USSR Academy of Sciences and directed it. Afterwards, I. Artobolevsky became its head of it. Mertsalov's main works concern the general theory of mechanisms, machines, theory of spatial mechanisms and thermodynamics.
In 1914–1916, N. Mertsalov published lithographic lectures on kinematics and the dynamics of mechanisms (Fig. 1.17). In "Kinematics of mechanisms" [16, 17] book he, like his precursors Yershov and Zernov, presented mechanisms analysis methods in

accordance with Willis's classification. The course included the basics of kinematics geometry, its applications concerning problems of mechanisms research based on the centrode method, swinging circle method, etc. Some problems of plane mechanisms synthesis were discussed (non-round wheels synthesis, motion transmission by centrodes and interrounding curves). To complement this part of applied mechanics, Mertsalov developed a set of problems the kinematics of plane mechanisms [18]. The consideration of spatial mechanisms was reduced to Hooke's joint consideration using Monge's projection method. In the 1920s Mertsalov delivered course of 20 lectures on the theory of spatial mechanisms for the Academy of Agricultural Sciences collaborators and conducted seminars to complement each of the lectures, a number of original analysis methods being developed by himself [19].

1.2.3.1 "Mechanisms Dynamics" by Professor N. Mertsalov [16]

Fig. 1.17: The title of lectures by N. Mertsalov "Mechanisms Dynamics" and "Kinematics of Mechanisms"

Mertsalov taught "Dynamics of Mechanisms" since 1909. In 1914, the lectures were organized and published by his disciple M. Felinsky, later a professor. This part was taught after "Mechanisms Kinematics" course and consisted of three parts: friction in machines, dynamic research of mechanisms, the stability motions of machines during operations. To go by the significance of the work, it should be remembered that until 1923 actually there was no systematic course for the dynamics of machines in world literature. Only in 1923, did Professor Wittenbauer's "Graphical dynamics" appear. In fact, Mertsalov's book was the first serious work on the fundamentals of dynamics of machines.

Four main primary methods of exposing the material can be highlighted: exposing from simple to complex, combination of an analytic method with graphic interpretation, extension of the analytic method (including elements of numerical methods) and its results to harder cases and their engineering interpretation. It has to be mentioned that the book could be the guide but did not give the strict sequence of concrete problem solving.

Before the Revolution, students had good schooling fundamentals, but basically no practical experience. It was easier for them to follow the scientific basis, but they had no idea of actual machines. That is why Mertsalov regularly completed his theoretic computations with basic constructional representations of these computations and (that was very important) gave substantial analysis of results and prognosis of the system’s conduct after changing some of the problem’s conditions.

The foundation of Part 1 (friction in machines) was the concepts of Poncelet and Coulomb’s law. The following topics were considered:

- Motion of a solid on a surface, including sloping, with the influence of positional forces.
- Motion by the action of spring, including the operation of auto-oscillations.
- Axis friction of a rocking pendulum.
- Friction on screw, wedge, higher pair, belt drive (Fig. 1.18).
- Friction of self-broken mechanisms and the efficiency of mechanism.
- Friction of swing and conditions of slipping.

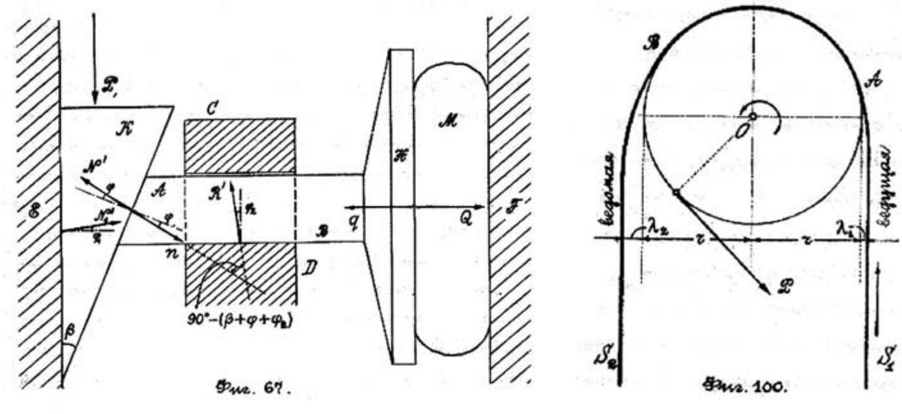


Fig. 1.18: Friction in mechanism of a wedge press and belt drive

To illustrate the method of teaching the subject of this part of the book, we’ll consider a fragment of the problem of the friction of a higher pair problem. Dry friction will be considered. Part of the basis of the reasoning about the nature of friction is a number of factors: the force of profiles interaction *Q,* their forms (curvature radii of contacting profiles – r_1, r_2), the kinematics of the transmission (p – the distance from the point of contact to the pole of gearing, thus giving in a non-obvious form one of the parameters of the speed of sliding), the way of interaction of the rubbed surfaces (coefficient of

friction – f). On the basis of Fig. 1.19, the analytic derivation of the elementary work of friction in an involute transmission is:

$$dA_{fr} = fQ^n p(1/ r_1+1/ r_2)\ ds. \tag{1}$$

Basing on this general case – non-involute transmission – is considered (Fig. 1.19 right); today we can present this as the multiplication of contact stress on the speed of sliding ($\sigma \cdot V_{sl}$).
There is reasoning about possible forms of wear (attrition) and the problem statement of even attrition on a profile and the selection of contacting a profile's section, ensuring minimal attrition.
In the second part (dynamic research of mechanisms) a mechanical system with forces depending only on the position is considered (positional system). To solve this problem the equation of motion is used in its energy form. Four fundamental requirements for the machine operating at periodic duty were offered:

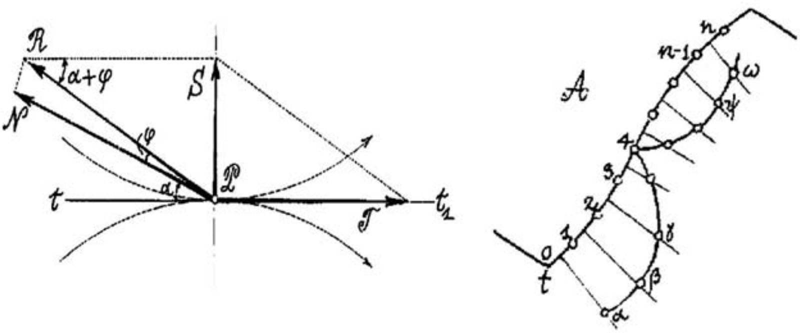


Fig. 1.19: Friction in a higher pair

- Strength of machine' s elements
- Sufficient stability on the foundation
- Evenness of a main shaft's movement
- The greatest efficiency in given conditions

On the basis of these requirements, the statements of the problems of dynamic research are given. Nowadays these problems are formulated in the following way:
Problem 1: Given: forces and initial velocity of any points of mechanism (at this part of the course Mertsalov considered only mechanisms with $W = 1$), to determine: the law of a mechanism's motion and the reactions in the kinematic pairs. This problem abuts onto the determination of the law of motion during acceleration-braking of a machine. Basically, the results of this problem give the answers to the first two requirements.
Problem 2: Given: forces to determine the conditions that would provide the cyclic rotation of a crank in given limits of angular velocity variation. That would be the answer to the third requirement.

The answer to the fourth requirement – determination of mechanism's efficiency – is formulated in a couple of lines which help us to understand that Mertsalov offered to evaluate losses caused by friction with the help of the sequential approximation method: at first to determine reactions of kinematics pairs without friction, then -losses caused by friction on the basis of the first part's material. Mertsalov used D'Alamber's principle Mertsalov used D'Alamber's principle (force's analysis), Lagrange's equation of second degree (design of dynamic model) and movement of an equation in energy form to solve these problems. The terms "reduced momentum of forces" and "reduced momentum of inertia" were absent. Wittenbauer came up with those terms much later and Smirnov used them in his "Kinetics..." [20].
Since Mertsalov considered only positional systems, he used the movement equation in its energy form for the solution of the initial problem of dynamics (problem 1). Problem 2 represented the set of two boundary problems: the problem of the maintenance of a mechanism's cyclic motion maintenance and the problem of a slider's angular velocity the variations of a given limits. It has to be mentioned that it was very hard for me to read this part. The reason were the extensive verbose explanations (nowadays they seem superfluous, however, I think they were necessary at the time) and divergence with modern terms and symbols. Presently, these problems are written down in a compact form, and something that used to be described in a number of pages can be put down in a number of lines. Unlike Wittenbauer's method, Mertsalov's method can now be presented as the first approximation of solution of the problem for the general case of system of the representation of a forces, meaning the forces that depend on position, velocity and time.
In the third chapter (stability motion of machines) the following questions were considered:

- Balancing of linkage
- Mechanisms with elastic links
- The problem of critical velocity of machines on foundation
- Balancing the flexible rotors

The problem of balancing a linkage was reduced to consideration of a crank-slider mechanism. The balancing, both with counterweight and crank-slider mechanism is examined. The problem concerned to reasoning about a machine's anchor to the foundation and influence of force of tightness of bolts, connecting the machine with its foundation, on the machine's stability. The question of the movement of mechanism with elastic links was examined in the example of the crank-slider mechanism, the slider of which was connected with a coupler by a cylindrical spring (Fig. 1.20).
First the simplified problem of the critical velocity of the crank's rotation, where the movement of point B was assumed to be harmonic and not dependent on the coupler's length, was examined. The dependence of position B on the crank's position was represented in the form of an harmonic series. It was shown that the accuracy of the computation's results was provided with the allowance of only the first two harmonics. Further, the influence of other harmonics and questions of the influence of friction were discussed. The paragraph ended with a recommendation on the calculation of springs and a number of methods for going through critical velocity. The question of critical velocities on the foundation was considered as a continuation of the previous question and was finished with a recommendation for designing foundations for slow- and fast-going machines.

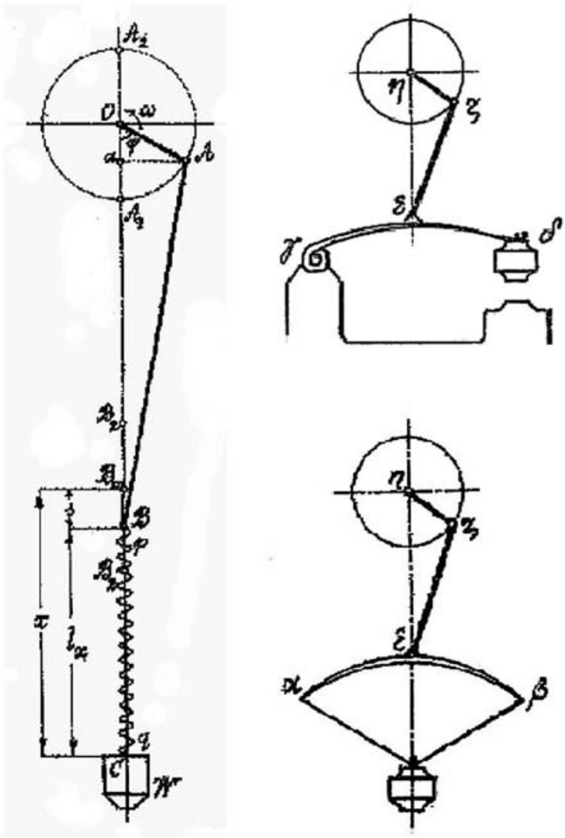


Fig. 1.20: Mechanisms with elastic links

In the conclusive paragraph balancing flexible shafts was examined. The mathematical approach to this problem's solution was simpler than previous. The essential part was discussion of practical methods of balancing: he gave the concept of the advantage of a mechanical system balancing over each element of it. To sum up the first two stages of the development of the Applied Mechanics course, it can be mentioned that the review of existing mechanisms was made a point of. It was the initial reason for purchasing existing collections. We can assume that mechanisms purchased by Yershov and, especially, by Orlov were the core of the IMTS and Moscow and Moscow University collections.

1.3. Semigraphical methods of Professor L. Smirnov (1929–1949)

Fig. 1.21. Professor L. Smirnov (1877–1954)

Professor L. Smirnov (FIg. 1.21), a student of Joukovsky just like Mertsalov, graduated from the IMTS in 1903 and was directed to wards a foreign mission for the perfection of the knowledge of noted scientists (Prandtl, Buh, etc.). After returning to Russia in 1906 L. Smirnov, with Mertsalov’s recommendation, stayed at the Applied Mechanics Department and also at the Department of Steam Engines. From that time on his life was closely connected with this university [13, 21]. From 1929 to 1949 he was Head of the Applied Mechanics Department (later, in the 1940s, the Theory of Machines and Mechanisms Department). In 1930 he founded and also headed the Department of “Steam Engines and Power Plants”. At the beginning of his scientific career he worked under Joukovsky’s supervision on the problems of the Theory of Regulation and Theory of Machines. It was he who was assigned to begin research concerning gearing to the engineer L. Reshetov (later a professor, and head of the TMM Department). This research resulted in the creation of unique gearing (for example a pinion with one tooth). In the 1940s, this research was continued by V. Gavrilenko, who later became a professor, Head of the TMM Department and who headed the science school of gearing at BMSTU. In the 1940s, under Smirnov’s supervision, the works concerning balance of rotors began at the TMM Department. These works were the basis for another scientific school at the BMSTU headed by Professor G. Petrov. During these years, besides manuals on kinematics and kinetics of machines and mechanisms, graphic-and-computational home-works on TMM, reference lists and teaching aides for these home-works were elaborated on and edited by Smirnov. Graphic procedures developed by Smirnov, including his unpublished works, were the foundations of a number of other technical university TMM courses. His influence on teaching the BMSTU course still remains. We can assume that it was in the 1930s, when on Professor Smirnov’s initiative, Reshetov’s works started the Soviet part of the collection of mechanisms.

The study of Mertsalov and Smirnov’s work proved that in general they had the same view on the subject of the Theory of Mechanisms. However, the contents and method of approaching the course were different in principle. What could be the reason for

disaccord of two scientists who basically had the same education and were brought up in the same environment? In opinion of the author of this chapter, the answer to this question, though he can be mistaken, is in political and social aspects.

In 1917, a great change in the political system of Russia happened. Until 1917, only well-to-do people who graduated from classical or applied sciences gymnasiums had a chance of entering higher educational institutions. Most of the Russian population workers and peasants - did not have that opportunity. The gymnasium graduates were well-trained in Mathematics, Physics, Chemistry, and other sciences. The process of teaching was developed according to their level. The courses were taught on a high theoretical level without going into the details about practical realization. Often material was given as ideas. After 1917, many workers and peasants were given the opportunity to get a higher education. However, their school education obviously wasn't as good: only a couple of years of schooling, end a weak natural sciences background. "Work faculties" (in Russian-RabFak) were organized to prepare them for studies in higher educational institutions. The most talented but grown-up people who had to combine work and studies were sent to these faculties. Professors had to teach at their level. Basically, the scientists had to switch to different methods of teaching with specific techniques of computation, in other words, to make an algorithm for all computations. That is what Smirnov, who was very fond of graphic methods, did. His love for graph-analytic methods was objectively explained.

At that time, a fast solution of technical problems with an accuracy sufficient for practical use, could be found only by graphic or graph-analytic methods. The main part of problems was reduced to a number of geometric transformations and scale computations with two arithmetic operations used – multiplication and division. For grown-up students, graduated from RabFak, that was easier than going deeper into the physical essence of a process. Here we also have to distinguish one more factor – a student's different life experiences before and after the revolution.

Students who entered technical universities after the revolution – Smirnov's group – had weak school backgrounds but good practical experience. They were locksmiths, blacksmiths, locomotive engine-drivers, mechanics, and so forth. They were familiar with the construction of machines, their exploitation and its results. That is why a substantial interpretation of representations was not as important as the strict technique of approaching the problem given. Of course, the class of problems being solved had to be decreased and the technique of a graph-analytic approach had to reach perfection. That very year, in 1926, his two manuals for higher schooling, "Kinematics of mechanisms and machines" [22] and "Kinetics of mechanisms and machines" [20] were published. Then, in 1932, Baranov's "Kinematics and dynamics of mechanisms" [23] course, which was based on the graphical methods developed by Smirnov (including the unpublished ones written by Baranov and Smirnov's lectures), was published. The books shouldn't be considered the only ones on applied mechanics. In that period, for example, the following books were published:

- D. Zernov, Applied mechanics
- A. Radzig, Applied mechanics
- D. Ruzsky, Kinematics of machines
- D. Ruzsky, General theory of machines
- L. Levenson, General theory of machines

- L. Levenson, Static and dynamics of machines
- L. Levenson, Kinematics of mechanisms
- V. Dobrovolsky, Dynamics of chains, etc.

1.3.1. “Kinematics of Mechanisms and Machines” [22]
The theoretical basis of the course was Ampère’s idea of kinematics as a form of geometry. Practicalities of the course came down to geometric constructions and introduction to all scale laws. The aim of teaching was to study the kinematics of mechanisms, which were most widely spread at that time. We can judge the contents of the book by its table of contents:

1. Trajectory, velocity and acceleration of a body moving rectilinearly
2. Curvilinear translational motion
3. Rotation of a body about a fixed axle

4–6. A plane moving relative to a parallel plane (three cases)

7. Acceleration center
8. Coriolis acceleration
9. Transfer from one plane to another by direct contact
10. Forced motion of train links, their velocity and acceleration

The first eight chapters concerned plane motion of a particle and body interpreted geometrically: the author constantly bound geometric image with mechanical quantity by scales.
The book was concluded with two chapters, were un analysis of the kinematics of a number of existing mechanisms was given, in, analysis of a machine mechanism’s model being forestalled by the design of the machine. You can get an impression of these chapter’s contents with the help of Figs. 1.22 and 1.23.

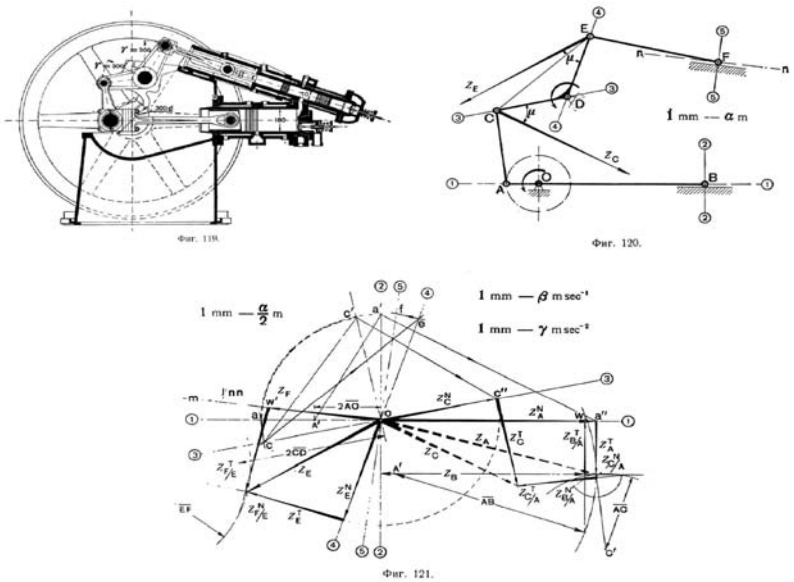


Fig. 1.22: Graphics research for the kinematics of Junker’s experimental engine

In Fig. 1.22 (kinematics: Figs. 1.19, 1.20, 1.21) a design of Professor Junker's experimental engine, its model and its velocity diagram are given. In Fig. 1.23 (kinematics: Figs. 1.22, 1.23) one can see the design of Wolseley's engine, its model and velocity diagram. The leverage here has two cranks and DOF = 2. The link of the cranks determines the mechanism's motion.

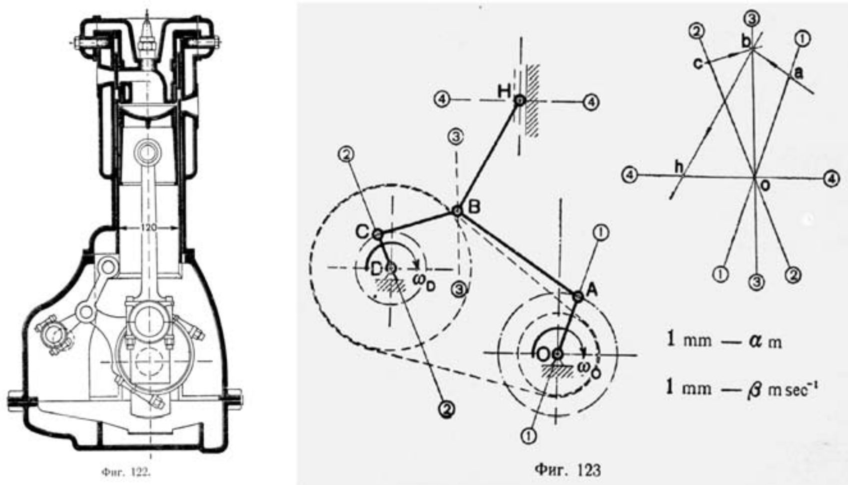


Fig. 1.23: Graphics research for kinematics of Wolseley's engine

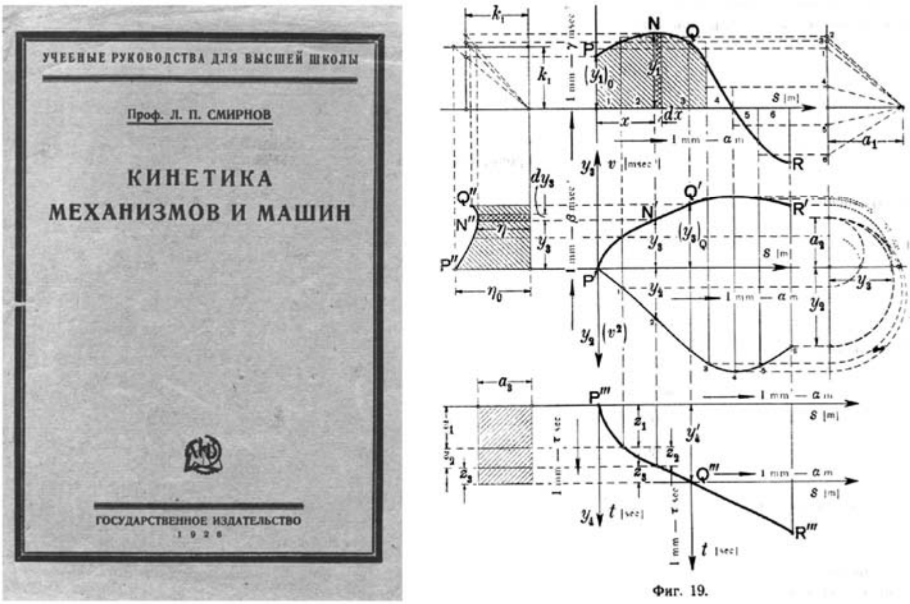


Fig. 1.24: The title of the text book by L.P. Smirnov "Kinetics of Mechanisms and Machines" and pages of graphics research of kinematics

1.3.2. “Kinetics of Mechanisms and Machines” [20]

It is to the point to quote a part of the introduction to “Kinematics…” and “Kinetics…” that gives us the idea of Smirnov’s credo: “… basing on 17 years of pedagogical experience, the author considered it necessary to put the so-called scales into all the formulas and graphic construction starting from the first pages … ”. The second quotation from the book’s introduction is “ …it had to be given the title “Mechanism’s Dynamics” but the author named it “Kinetics of Mechanisms and Machines” (Fig. 1.24), meaning title to emphasise the dominating part that the study of the capacity of kinetic energy plays as the result of the work of applied forces”.

The book consists of ten chapters. Subjects considered in this book were closely connected with Mertsalov’s book. There were changes in the sequence of themes, the questions of machines’ stability during movement were missing, but the concept of “reduced moment of forces” and “reduced moment of inertia” was introduced. It was the subject-matter of these themes that was fundamentally different.

In the first and the second chapters, graphic and experimental methods (Fig. 1.25) of simple movement’s research were given. In Chapters 3 and 5, static and dynamic balancing of rotors was considered. Graphic and experimental methods of determining any detail’s moment of inertia were shown in Chapter 4 (Fig. 1.26).

The last four chapters concerned the movement of mechanism. Unlike Mertsalov, Smirnov used Wittenbauer’s terms of “reducing” forces and masses of mechanisms.

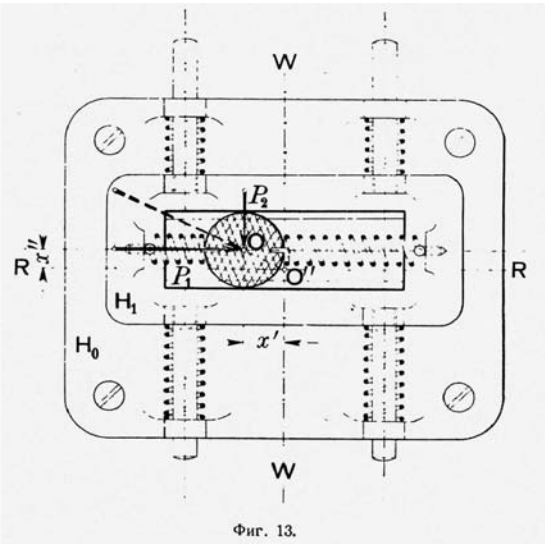


Fig. 1.25: Experimental installation for studying the movement of a solid body under the action of elastic forces

Just like Mertsalov, Smirnov worked on the dynamics of conservative systems and therefore used the equation of motion in the energy form. He used Wittenbauer’s method for determining a flywheel’s moment of inertia. It proved one more time his passion for graphic methods. It could be said that graphic methods were methods of

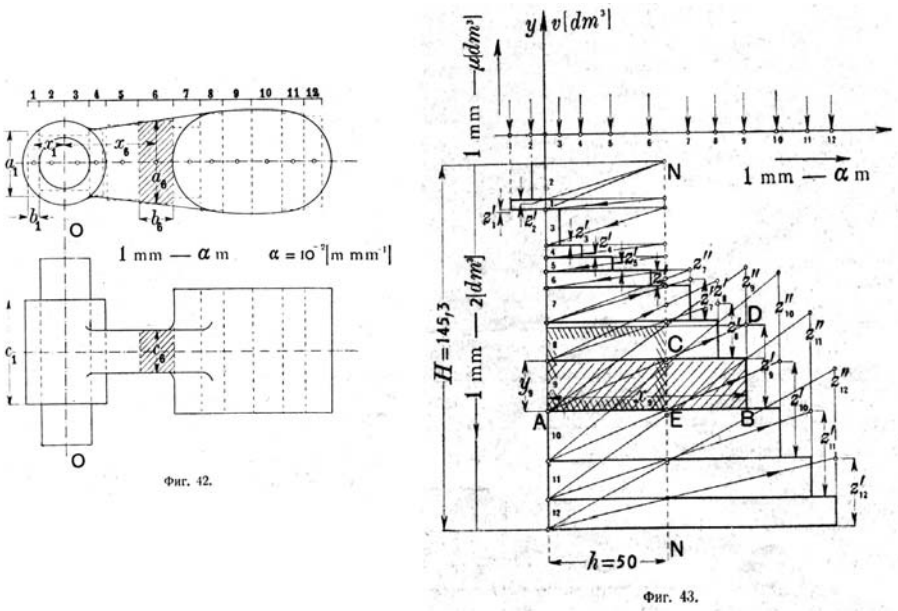


Fig. 1.26: Graphical method of the determination of the moment of inertia

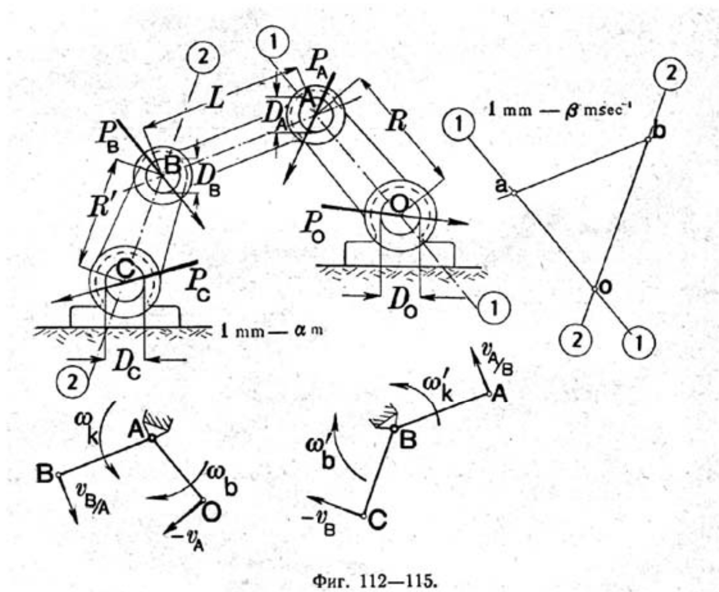


Fig. 1.27: Method of determination of friction losses in linkages

explanation for Mertsalov, while they were methods of calculation for Smirnov. Introducing the term “reduced force” into the course allowed Smirnov to show the method of determination of friction losses in a linkage – in fact, it was the explanation of the sequential approximations method for efficiency used by Mertsalov (Fig. 1.27). In comparison with the method of “friction circles” from a later published books and suitable only for simple mechanisms, this new approach was much more modern. We can judge the contents of material concerning a higher pair by two chapters of the book [23]: “Transmission of rotary motion by gear wheels” and “Gearings”. One can note that the material of the chapters was practically oriented. The former one was concluded with an involute gearing estimation in respect to a transmission’s life and manufacturability, and also with power loss and the transmission’s performance evaluation methods carried out by purely geometric methods (Fig. 1.28). In the latter triangle of velocities method for gearings, planetary mechanisms (including differentials) kinematics analysis was introduced. Besides, technological and functional features and assembly questions were considered (Fig. 1.29).

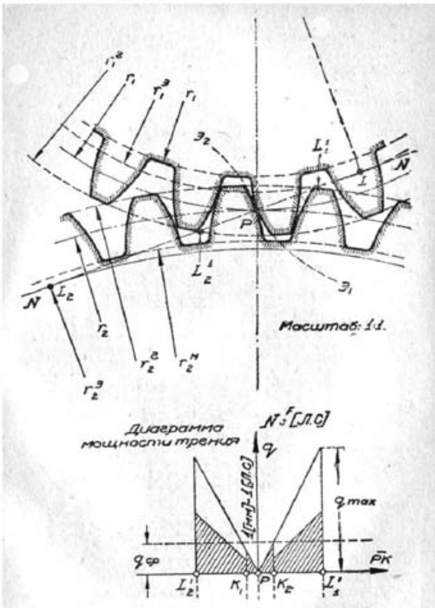
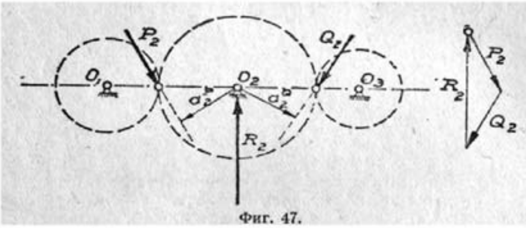


Fig. 1.28: The title of the text-book by G. Baranov “Kinematics and Dynamics of Mechanisms” and pages for the frictions of gearing

Smirnov developed “Applied Mechanics Reference Lists and Metodical” [24, 25] on the basis of his own books and lectures, Reshetov, Baranov and Shitikov (who would later establish Kazan’s TMM school) taking part in its preparation.

At the end of the 1920s and in the 1930s, young people who had finished normal school courses and had a good grounding entered the higher educational institutions (institutes and universities) of our country. The fact made it possible to raise the level of the applied mechanics course. At the end of the 1930s, the Applied Mechanics Department

introduced graphic- and -computational home-work. They were prepared by the department’s member N. Vzorov under Smirnov’s direction and published in 1940. The works concerned:



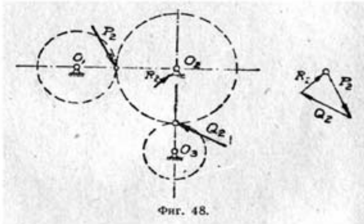


Fig. 1.29: About gearing configuration

- Absorber for automatic coupler of Peerless system research
- Involute gearing of spur gears research
- Planetary reducer with spur gear research
- Kinematics research of rectilinear motion (graphical integration and differentiation)
- Kinematics research of Junker’s system mechanism
- Cam kinematics research by Gartman’s method
- Determination of a flywheel’s moment of inertia
- Balancing of rotating masses

1.4. Smirnov’s pedagogic school

1.4.1. Professors L. Reshetov, A. Savelova, V. Gavrilenko, N. Skvortzova (1949–1978)

Professor L. Smirnov was the last head of the Department of «Applied Mechanics», who got higher education before the Revolution. In the 1940s, the 20th-century department was renamed to the Department of the «Theory of mechanisms and machines». L. Smirnov four disciples one by one were the head of the Department. They are Professors L. Reshetov and V. Gavrilenko, Ass. Professors A. Savelova and N. Skvortzova. They received their higher education during Soviet rule.

Professor Smirnov’s love of semigraphical methods and great practical work in the field of Applied Mechanics and steam engines left an imprint on the formation of a training school. On the other side, the school’s formation passed against the background of public and political events in the country, such as industrialization, Stalin’s terror, the Second World War, the fight against cosmopolitanism, persecution of genetics.

The pre-revolutionary era’s child, a disciple of N. Joukovsky, he and his predecessors, believed that the discipline should meet the demands of modern technology. Therefore the course is to be dynamic, using modern science and technology achievements to deal with current challenges. The attributed a special place in the teaching activities to computing techniques. This meant that computational procedures led strictly to formal graphical reconstruction and operations. The second could be found by two mechanical actions, multiplying and dividing. It was convenient before, although it is use-less and even harmful today!

Points of Smirnov’s methodical systems [21]:

1. 2-hour class material was kept during the classes. For this purpose his pupil L. Reshetov developed a design for the boards and the BMSTU workshops manufactured them.
2. Classes were accompanied by not only the story, but also show material models, in the cinema and on film. The lecture halls were so equipped.
3. For a successful study of the course, the separate sections were presented in the form of information lists and “blind” drawings, which the students inserted into note-books. For the same purpose, a series of teaching manuals and large number printed benefits were produced and are now stored in the BMSTU library.
4. Passing the course must be accompanied by laboratory classes using devices, which were designed and made by the workshop of the Department or BMSTU.

Implementation of this program led to the change of the content and methods of lectures, exercises, examinations and tests. The mission took many years and was completed after the death of the teacher.

In the prewar years, under the leadership of Smirnov, collaborators at the Department made, to the orders of industry, a number of activities which student courses included. Reshetov’s study theory of involute cog-wheel correlation, and the design of cam mechanisms.

V. Gavrilenko introduced some of the tasks of dynamics in the style of Tolle; a graphic method of designing techniques for artillery and tank differentials; processing of design cog-wheels; use of the methods of TMM in the study of movement of fuel in the fire-box of a locomotive.

A. Savelova (was invited to join Smirnov’s Department in 1936): work on balancing techniques and analysis balancing machines, the results of which served as the basis of the reading TMM course of this section, and for laboratory exercises with the students.

Professors Reshetov and V. Gavrilenko made the largest contribution to the development of the course. All their lives, their fortunes, triumphs and failures were connected with the department [26].

Gavrilenko was the son of the first democratically elected director, Alexander Gavrilenko, a BMSTU Professor, who was elected director of BMSTU in 1905. In 1914 the First World War began. At that time, V. Gavrilenko was a gymnasia pupil. In 1917, on graduating from high school, he entered the Mihaylovskoye Military School. When the Great October revolution of the Russian people took place, officers didn’t generally take to the power “Bolsheviks” and the civil war began. Young people were evacuated from Petrograd, and joined the so-called volunteer army, which fought against the Red Army. The civil war ended in victory for the Red Army and the remaining volunteer army emigrated Vladimir Gavrilenko emigrated from Russia too. After a few years, he decided to return to Russia. The chairman of the Higher Council of the National Economy, Bogdanov, cooperated in this migration, he was a student of the University. Vladimir returned trough the instrumentality of Bogdanov. A significant factor in that was that Bogdanov respected Alexander Gavrilenko, the director. The fact is that A. Gavrilenko was different that he, generally speaking, did not quite let go of power, in particular the police, to the BMSTU territory, although he was rather more right-wing than left-wing, socialist or socialist-revolutionary. In 1925, V. Gavrilenko returned to Russia and entered BMSTU. To survive, he worked first as an assistant and then a

locomotive driver. In the 30s year, he defended his diploma project on the theme: commercial locomotive of series "E" and became a research worker at the Institute of Reconstruction of the Steam Traction.

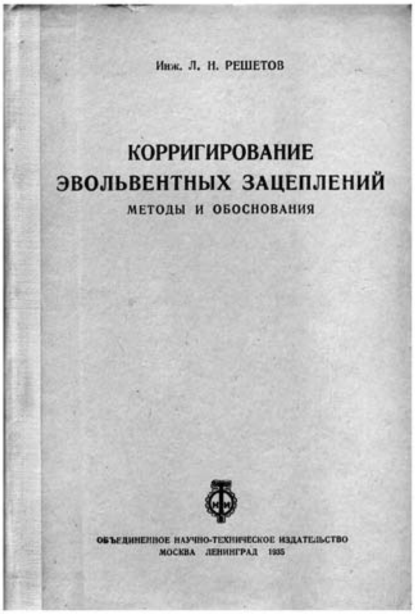
Fig. 1.30: Professor Vladimir Gavrilenko

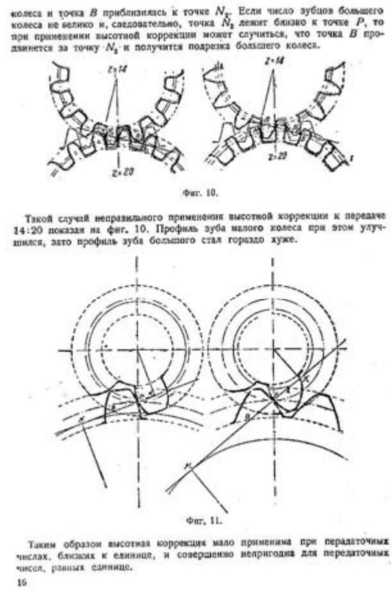
In the 1930s the first five-year plan began, and at the same time general reconstruction literally began, including the railways. Heavy-duty steam locomotive was required instead of the small steam locomotives and steam of mean power, which were bought in the United States. That steam had the so-called stoker heating – a screw took the coal from the tender to the fire-box. The trouble was that the efficiency was very low because of incomplete combustion of the fuel. Gavrilenko made two improvements to the stokers, which increased the cost of the heating of the stoker. In addition, he offered a the method of cleaning the boiler. For that, he was awarded a premium, 75% of which he gave to the Country's Defense Fund, and with the remaining 25% he bought a new bicycle. In the 1932 Vladimir Gavrilenko (Fig. 1.30) received an invitation from Professor Leonid Smirnov to join the department and at first he was a moonlighter.
In 1937, Gavrilenko was arrested. Perhaps it was an echo of the Civil War. He was interned in the so-called Samarlag (Samara camp), involved in the construction of the Kuybyshevskoi Hydroelectric Power Station. As he had a higher education and was a well-known specialist, he worked in the technical corps.
He was released from Samarlag at the end of 1938. Smirnov gave, a very good description of Gavrilenko as an engineer and, as a specialist, "regardless of Stalin". At the end of 1939, he returned to teaching and research activities. At that time he was interested in the design of gears. He said that he "was surprised that a train of gears theory was lacking although there were the systems of correlation adjustment of Bakengem, Schroeder and others." This infatuation became his life's work: he developed a geometric theory of gears. This work was carried out during the Great Patriotic War. He remembered that his continuing wish to eat and deeply cold winters followed him. Cold was such that people of the BMSTU didn't take off coat. In his winter coat, Vladimir Gavrileko originated his geometrical theory of gears.

L. Reshetov (Fig. 1.31) entered BMSTU in the same year that Gavrilenko did. In 1930 he graduated from the BMSTU and joined the Department of TMM. At the suggestion of Smirnov, he focused on analyzing existing designs and the manufacture of gears. One of his first works was the comparative analysis, in which he proposed his vision of this problem [27] (Fig.1.32). At the same time, he focused on the design of a cam grinding machine [28] (Fig.1.33). This small work was subsequently significantly expanded [29]. In his work, he was closely connected with industry, especially the railways. In the 1950s, L. Reshetov became seriously interested in the problem of the structure of mechanisms.

Fig. 1.31: Professor L.N. Reshetov

This enthusiasm turned to serious study and the creation of the original design theory of mechanisms without excess links [30, 31]. Such mechanisms he described as "rational". Reshetov is the author of more than 50 inventions, and much of it was implemented. His theoretical work was usually accompanied by the creation of models or patterns. He made many models himself. He made preliminary sketch models from details of the meccano sets. Thanks to his work, the collection of models was significantly enriched with new models of the various mechanisms, many of which are unique.
Friendly relations were between Reshetov and Gavrilenko, while lasted they worked together in the same department. Then in the late 1930s, Reshetov left BMSTU and held the department of TMM in the Automotive Institute. In 1949, Smirnov died. A member of the Communist Party, Ass. Professor Alexandra Saviolova (Fig. 1.34) temporarily became the head of the department. Exactly at that time, a cycle of laboratory work was carried out, much of which is still used now, but with modern materials and a computer base. In the same years the section of mechanisms with elastic units disappeared from the lectures.
In 1951 Professor Reshetov joined the department of TMM at BMSTU. At other times, the department would suggest to V. Gavrilenko, but he had the stamp of the "white officers". It was at that time when series of lectures on all sections of the course was written and a compilation of complex tasks of course project was prepared.




колеса и точка B приблизилась к точке N_2. Если число зубцов большего колеса не велико и, следовательно, точка N_2 лежит близко к точке P, то при применении высотной коррекции может случиться, что точка B продвинется за точку N_2 и получится подрезка большего колеса.

Фиг. 10.

Такой случай неправильного применения высотной коррекции к передаче 14:20 показан на фиг. 10. Профиль зуба малого колеса при этом улучшился, зато профиль зуба большого стал гораздо хуже.

Фиг. 11.

Таким образом высотная коррекция мало применима при передаточных числах, близких к единице, и совершенно непригодна для передаточных чисел, равных единице.

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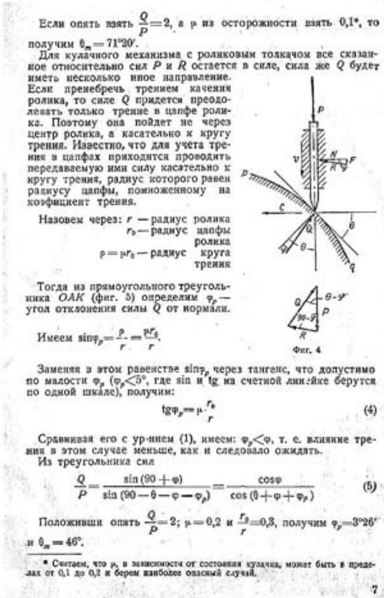
Fig. 1.32: The cover of Reshetov’s book “The correction of involute toothing” with an example page [27]


Если опять взять $\frac{Q}{P}=2$, а μ из осторожности взять 0,1*, то получим $\theta_m=71°20'$.

Для кулачного механизма с роликовым толкачом все сказанное относительно сил P и R остается в силе, сила же Q будет иметь несколько иное направление. Если пренебречь трением качения ролика, то силе Q придется преодолевать только трение в цапфе ролика. Поэтому она пойдет не через центр ролика, а касательно к кругу трения. Известно, что для учета трения в цапфах приходится проводить передаваемую ими силу касательно к кругу трения, радиус которого равен радиусу цапфы, помноженному на коэфициент трения.

Назовем через: r — радиус ролика
r_0 — радиус цапфы ролика
$\rho=\mu r_0$ — радиус круга трения

Тогда из прямоугольного треугольника OAK (фиг. 5) определим φ_p — угол отклонения силы Q от нормали.

Фиг. 4

Имеем $\sin\varphi_p=\frac{\rho}{r}=\frac{\mu r_0}{r}$.

Заменяя в этом равенстве $\sin\varphi_p$ через тангенс, что допустимо по малости φ_p ($\varphi_p<5°$, где sin и tg на счетной линейке берутся по одной шкале), получим:

$$\operatorname{tg}\varphi_p=\mu\cdot\frac{r_0}{r} \quad (4)$$

Сравнивая его с ур-нием (1), имеем: $\varphi_p<\varphi$, т. е. влияние трения в этом случае меньше, как и следовало ожидать.

Из треугольника сил

$$\frac{Q}{P}=\frac{\sin(90+\varphi)}{\sin(90-\theta-\varphi-\varphi_p)}=\frac{\cos\varphi}{\cos(\theta+\varphi+\varphi_p)} \quad (5)$$

Положивши опять $\frac{Q}{P}=2$; $\mu=0{,}2$ и $\frac{r_0}{r}=0{,}3$, получим $\varphi_p=3°26'$ и $\theta_m=46°$.

* Считаем, что μ, в зависимости от состояния кулачка, может быть в пределах от 0,1 до 0,2 и берем наиболее опасный случай.

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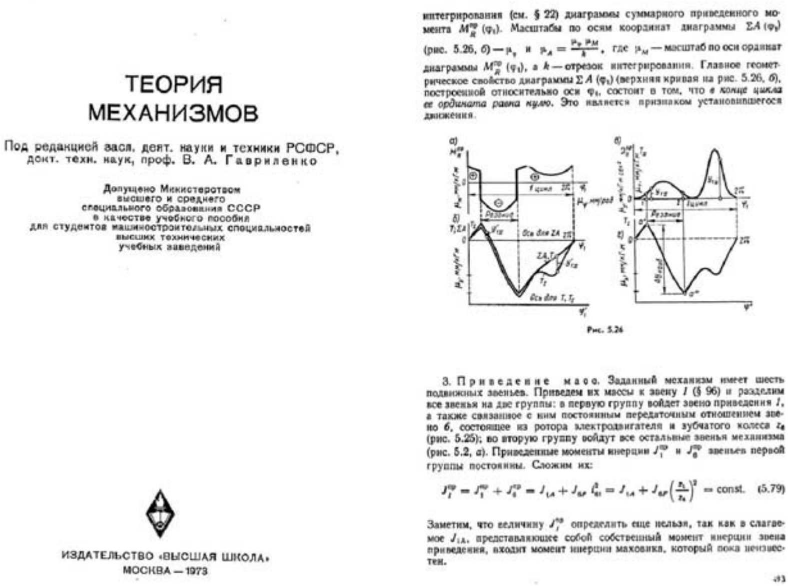
Fig. 1.33: The cover of Reshetov’s book “The cam mechanisms of automatic lathe” with an example page [28]

Fig. 1.34: Ass. Professor Alexandra Saviolova



ТЕОРИЯ
МЕХАНИЗМОВ

Под редакцией засл. деят. науки и техники РСФСР,
докт. техн. наук, проф. В. А. Гавриленко

Допущено Министерством
высшего и среднего
специального образования СССР
в качестве учебного пособия
для студентов машиностроительных специальностей
высших технических
учебных заведений

ИЗДАТЕЛЬСТВО «ВЫСШАЯ ШКОЛА»
МОСКВА — 1973

интегрирования (см. § 22) диаграммы суммарного приведенного момента $M_{R}^{пр}(\varphi_1)$. Масштабы по осям координат диаграммы $\Sigma A(\varphi_1)$ (рис. 5.26, б) — μ_φ и $\mu_A = \frac{\mu_\varphi \mu_M}{k}$, где μ_M — масштаб по оси ординат диаграммы $M_{R}^{пр}(\varphi_1)$, а k — отрезок интегрирования. Главное геометрическое свойство диаграммы $\Sigma A(\varphi_1)$ (верхняя кривая на рис. 5.26, б), построенной относительно оси φ_1, состоит в том, что *в конце цикла ее ордината равна нулю*. Это является признаком установившегося движения.

Рис. 5.26

3. Приведение масс. Заданный механизм имеет шесть подвижных звеньев. Приведем их массы к звену *1* (§ 96) и разделим все звенья на две группы: в первую группу войдет звено приведения *1*, а также связанное с ним постоянным передаточным отношением звено *6*, состоящее из ротора электродвигателя и зубчатого колеса z_6 (рис. 5.25); во вторую группу войдут все остальные звенья механизма (рис. 5.2, а). Приведенные моменты инерции $J_1^{пр}$ и $J_6^{пр}$ звеньев первой группы постоянны. Сложим их:

$$J_I^{пр} = J_1^{пр} + J_6^{пр} = J_{1A} + J_{6P}\, i_{61}^2 = J_{1A} + J_{6P}\left(\frac{z_1}{z_6}\right)^2 = \text{const.} \quad (5.79)$$

Заметим, что величину $J_I^{пр}$ определить еще нельзя, так как в слагаемое J_{1A}, представляющее собой собственный момент инерции звена приведения, входит момент инерции маховика, который пока неизвестен.

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Fig. 1.35: The textbook cover page and an example page of part dynamic

Subsequently with these notebooks textbook of TMM was established by members of the department under the direction of V. Gavrilenko.

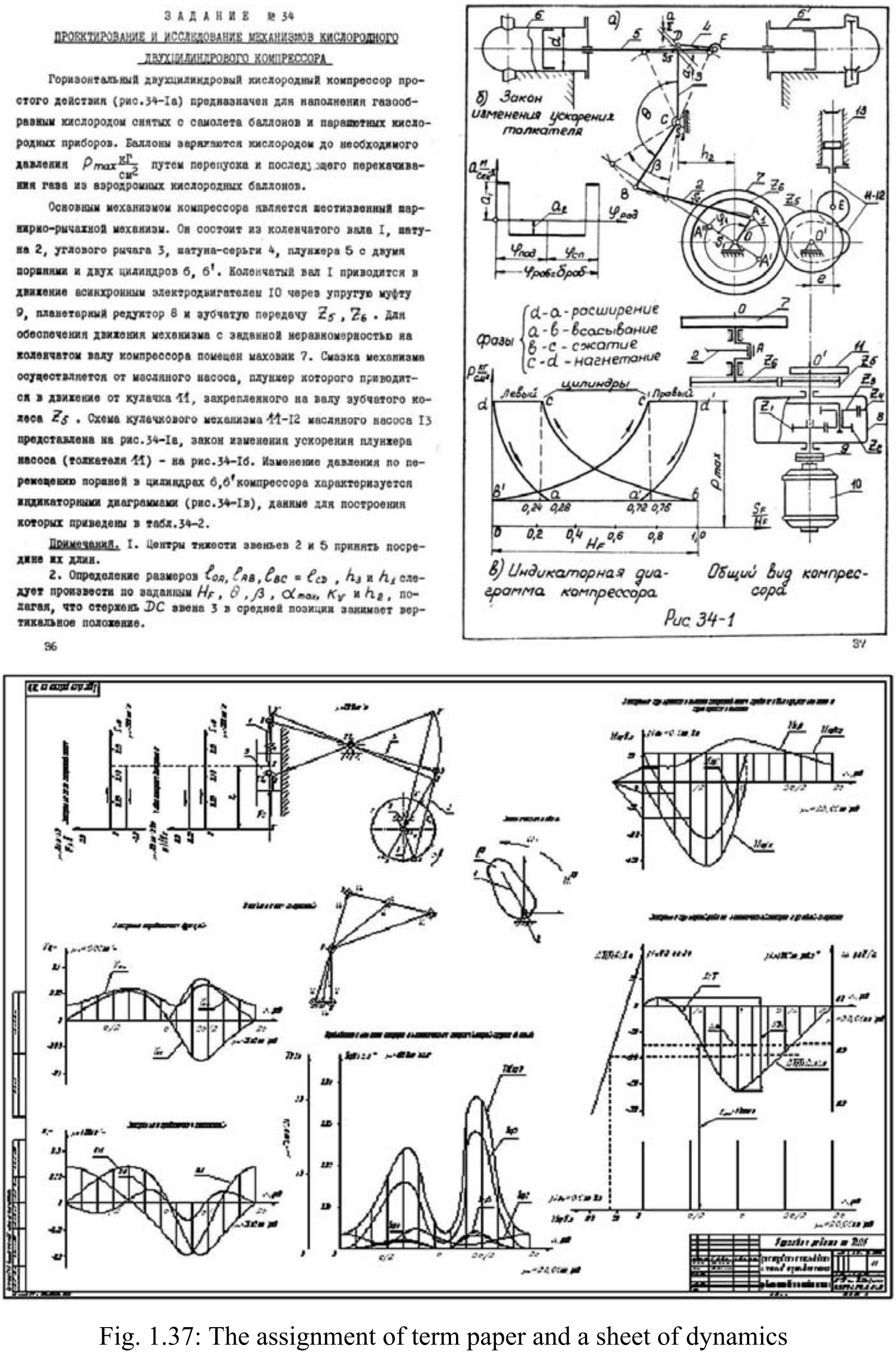
In 1962, for a variety of reasons, the Department of TMM was going to close and Re-shetov could not remain as the Head of the Department. The Communist Party’s group at the department offered the seat of the head of the Department to Professor V. Gavrilenko and so the department was saved from closure.

In the 1960s, the Department had a large group of young professors and researchers. Most of them worked under the direction of Professor V. Gavrilenko and his disciple, Ass. Professor Nika Skvortzova (Fig. 1.36). The system of calculation of gears, which was proposed to Gavrilenko, entered the State standard. In it 1966, under the direction of Gavrilenko, a scientific group was formed, whose task it was to lead scientific research and experimental research work of crank-type, planetary, and wave gears. In this group, based on the theory of an involute gear developed by Gavrilenko, special gear drives with an internal involute gear and wave gears were designed. The results of this team, and also the model established by the department's students as the results of the scientific work, are to be found in the collections of the department. In 1973, Professor Gavrilenko became a member of to the IFToMM Permanent Commission on Education.

Fig. 1.36: Ass. Professor N. Skvortzova

During those years, the TMM course was significantly enhanced. In connection with the member of Gavrilenko passion course for the train of gears, this section was expanded. A major achievement of that time was a textbook of TMM [32] (Fig. 1.35), in which many professors took part, and also a series of tasks for a comprehensive course project (Fig. 1.37). The project required knowledge of six themes. The themes were:

- Elementary metric synthesis of linkage
- Dynamic research of a machine unit with one degree of freedom
- Power calculation of a linkage
- Design machine-tool linkage and gearing based on the method of Gavrilenko
- The selection of the number of teeth of the elementary schemes of planetary mechanisms with one degree of freedom
- Designing a cam mechanism using set cyclic schedules and an admissible pressure angle



ЗАДАНИЕ № 34

ПРОЕКТИРОВАНИЕ И ИССЛЕДОВАНИЕ МЕХАНИЗМОВ КИСЛОРОДНОГО ДВУХЦИЛИНДРОВОГО КОМПРЕССОРА

Горизонтальный двухцилиндровый кислородный компрессор простого действия (рис.34-Iа) предназначен для наполнения газообразным кислородом снятых с самолета баллонов и парашютных кислородных приборов. Баллоны заряжаются кислородом до необходимого давления $\rho_{max} \frac{кГ}{см^2}$ путем перепуска и последующего перекачивания газа из аэродромных кислородных баллонов.

Основным механизмом компрессора является шестизвенный шарнирно-рычажной механизм. Он состоит из коленчатого вала I, шатуна 2, углового рычага 3, шатуна-серьги 4, плунжера 5 с двумя поршнями и двух цилиндров 6, 6'. Коленчатый вал I приводится в движение асинхронным электродвигателем IO через упругую муфту 9, планетарный редуктор 8 и зубчатую передачу Z_5, Z_6. Для обеспечения движения механизма с заданной неравномерностью на коленчатом валу компрессора помещен маховик 7. Смазка механизма осуществляется от масляного насоса, плунжер которого приводится в движение от кулачка 11, закрепленного на валу зубчатого колеса Z_5. Схема кулачкового механизма 11-I2 масляного насоса I3 представлена на рис.34-Iа, закон изменения ускорения плунжера насоса (толкателя 11) - на рис.34-Iб. Изменение давления по перемещению поршней в цилиндрах 6,6' компрессора характеризуется индикаторными диаграммами (рис.34-Iв), данные для построения которых приведены в табл.34-2.

Примечания. I. Центры тяжести звеньев 2 и 5 принять посредине их длин.

2. Определение размеров ℓ_{OA}, ℓ_{AB}, $\ell_{BC} = \ell_{CD}$, h_3 и h_1 следует произвести по заданным H_F, θ, β, α_{max}, K_V и h_2, полагая, что стержень DC звена 3 в средней позиции занимает вертикальное положение.

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Fig. 1.37: The assignment of term paper and a sheet of dynamics

In 1977, after the death of Professor Gavrilenko and up to 1978, the Department was led by Ass. Professor Nika Ckovrtzova. At that time, a laboratory of TMM was established in one of the branches of the BMSTU.

1.5. Recent times

Since 1978, Academician of the Russian Academy of Sciences Professor Konstantin Frolov is the head of the BMSTU Department of TMM (Fig. 1.38). He also started a restructuring of the course. The section of dynamics was returned to the course in consideration of the elasticity of units. Section of vibroprotection and the design basis of a robot's schemes were introduced. The TMM textbook was redesigned and included several publications [33–35]. The laboratory cycle was also modernized.

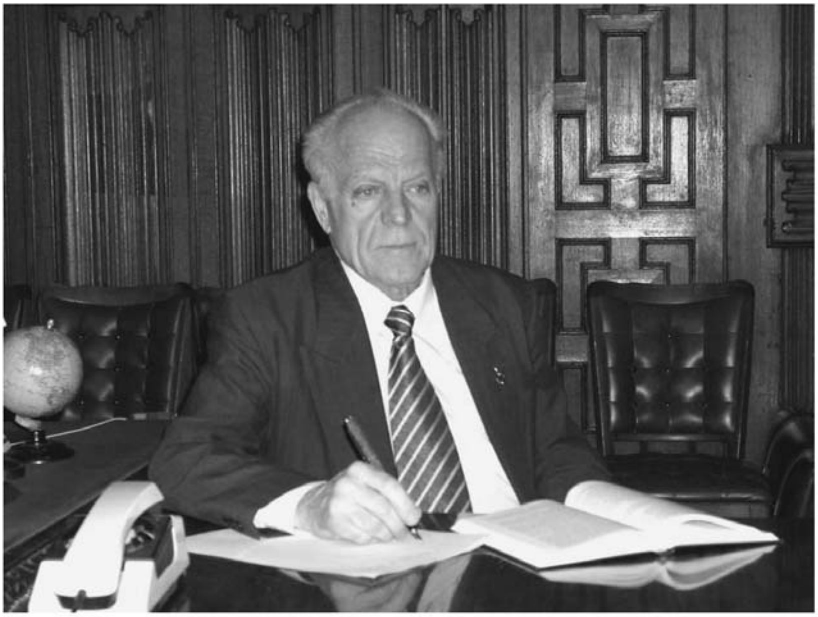


Fig. 1.38: Academician of the Russian Academy of Sciences Professor Konstantin Frolov

Besides the basic program task-oriented programs of the course of the TMM course have been created. Professor I. Leonov has developed a program for the faculty of "Engineering business and management". Professor A. Golovin has developed a program for students of technological specialities in which, as against the traditional program, the essential attention is given to the properties of mechanisms. This program has been maintained by books on designing multibar linkages [36] and on dynamics of mechanisms [37]. Under the direction of Ass. Professor V. Tarabarin, the laboratory's of the practical work on the TMM [38] has been advanced and the teaching and methodical complex TMM has been created in an Internet network (tmm-umk.bmstu ru), electronic tutorial at the course of the TMM [39] has been developed. The cover of the book [37] and titular pages of the website is shown in Fig. 1.39.

The most capable students gained the possibility of carrying out make special tasks,containing research elements and the solution of some scientific and practical tasks of the TMM instead of the standard course. The works of some students have been published in the well-known Soviet and Russian scientific and technical publications. In recent years (2000–2006), several students took part in the international conferences in Italy, Romania and Austria.

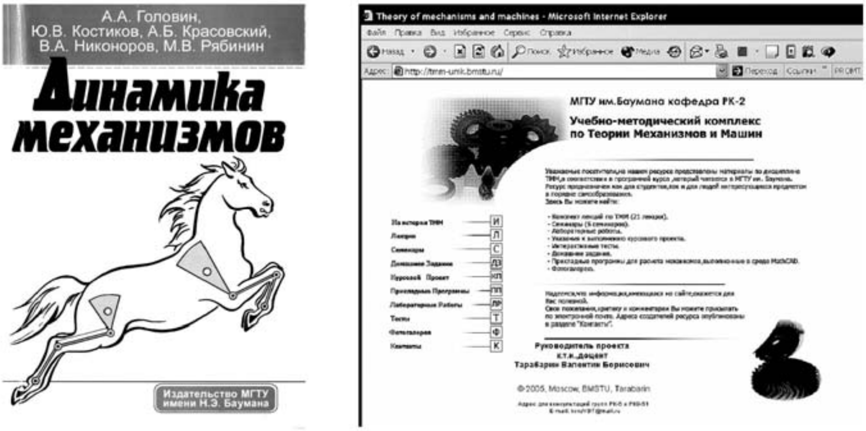


Fig. 1.39. Cover of the books [37] and titular pages of the website

The collaborators of the TMM Department became members of Committees and Commissions of the IFToMM (Professor A. Golovin, of the Permanent Commission on the MMS History in 1999, of the Committees on Linkages and Cams and Education in 2007; Ass. Professor V. Tarabarin of a Permanent Commission on the history of MMS

Fig. 1.40: The conferees of the Third International Workshop on the History of Machines and Mechanisms (BMSTU, Moscow, 17–19 May 2005)

in 2007). The collaborators of the Department regularly participate in international events in Italy, Finland, Romania, Germany, Austria, China, USA, and in return, regularly foreign scientists.

In 2005, for the first time in its history, the Department held an International Conference on the MMS History, with the participation of scientists from 11 countries (Fig. 1.40). Particular attention is given to the work of a collection of mechanisms (see Chapter 2).

2. HISTORY OF APPLIED MECHANICS CABINET AND MECHANISMS COLLECTION

2.1. The first models and start of collection

The scientific and technical revolution which began in the 18th century, led to society becoming aware of the need for training engineers. The first such school was École Polytechnique, founded in 1794 in Paris by G. Monge. In the 19th century, high technical institution and engineering universities began to appear. In Russia the first such institution was the Institute of Corps Engineers of Communications (today – Petersburg University of Transport) opened in 1810 in St. Petersburg. A. Betancourt took great part in its establishment.

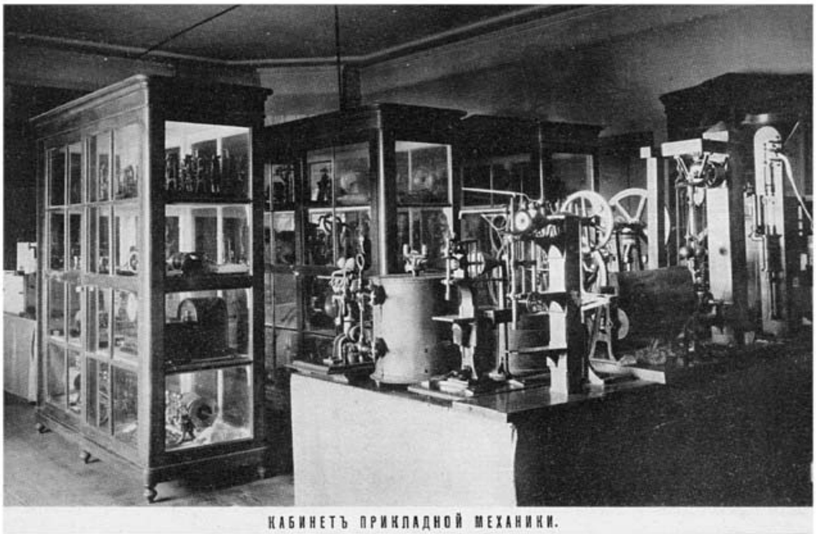


Fig. 2.1: A cabinet of "Applied mechanics" at the 19th beginning of century (the photo from museum of BMSTU)

Naturally, physical models of machines and mechanisms were accepted and effective means of engineering education. All major universities and technical institutes had significant collections of such models. Collecting, designing and manufacturing of the models of machines and mechanisms were practiced by A. Betancourt (Royal Cabinet in Spain in the period 1791–1808), J.F. Redtenbacher (Polytechnic Institute in Karlsruhe in the period 1840–1863), J. Schröder (Polytechnic Institute in Darmstadt), F. Reuleaux (Berlin Higher Technical College during the period 1879–1905), F. Orlov (Moscow University and the Imperial Moscow Technical Secondary School during the period 1872–1892), I. Rachmaninoff (Kiev University during the period 1853–1883), V. Ishmenitsky (Kharkov University during the period 1872–1882) and many other scientists. The largest collection of kinematic models (more than 800) was collected by

A. Golovin and V. Tarabarin, *Russian Models from the Mechanisms Collection of Bauman University*.

F. Reuleaux. This collection was considered to be ideal for technical schools in Germany and other countries [40–42]. Unfortunately, most of the collection was lost in 1945, during the bombing raids.

In Russia many of the college and university collections were lost in the 1930s–1970s a misunderstanding of the role of the old technical solutions in the training of students. Besides, due disregard of the Soviet officialdom to the technical social history, shîuld also be noted. For this reason to difficulty appeared in distributing of the collections and there was a lack of resources to keep them in decent condition. Some universities could not keep the old models, but also did not replenish their collections with new models. One of the world's largest collections of models is a collection at the Bauman University TMM Department. At present, the collection comprises more than 600 models.

Fig. 2.2: The first models of a cabinet of applied mechanics

A cabinet of mechanisms at the Department of Applied Mechanics (Fig. 2.1) was established in the mid-19th century. The collection consisted of models bought in Germany from the collections of J. Redtenbacher, J. Schröder and F. Reuleaux. Some models of the collection date back to the years 1861–1862 (Fig. 2.2). The first model was the original orthogonal spatial gear transmission. Input wheel of the transfer is a flat wheel on the front of which is only one tooth, made like Archimedes spiral. The inscription "Ivan Naumov, 1861" was engraved on the model. The second model was centrifugal regulator with an inertial element as a massive ring. On the bases of a model tablet was the inscription "Pavel Ivanov, 1862". These models visually coincide with the drawings catalogue [43], although they have small design differences. There are two sources from which these models could have reached the collection of the Cabinet of Applied Mechanics. Either they were bought in Germany (and the engraved names show people financing their purchase), or they were produced in the workrooms of

IMTS from the drawings catalogue [43]. Unfortunately, most of the archives of IMTS were destroyed during the Second World War and, therefore, it is impossible to find documents substantiating or disproving these assumptions. We can assume that Yershov established his collection sometime after 1843, when he began to teach practical mechanics and descriptive geometry in the senior class of the Moscow 3rd gymnasium, opened in 1839. Dates on the saved models belong to the period of the work of Ershov at the IMTS. It suggests that the Redtenbacher model was also bought the years of the work of Yershov, and the Cabinet of Applied Mechanics was purchased in the mid-19th century and included more than one hundred models.

Fig. 2.3: Cases with Reuleaux's models which were saved in the collection of BMSTU

In 1872, Fiodor Orlov was head of the Department of Applied Mechanics on the appointment of the Council of IMTS. During his mission in 1870–1872, he was impressed by the methods of models demonstration used in the teachings of Reuleaux. When he returned to IMTS he paid great attention to the Cabinet of Applied Mechanics, and broadened and systematized the collection of mechanisms. He took part in the creation of Cabinets of Applied Mechanics in most universities of Russia. The IMTS collection at that time was enriched with models created by Reuleaux and manufactured in Berlin in G. Voigt's workrooms. Figure 2.3 shows the cupboards from the Cabinet of Applied Mechanics with the models of Reuleaux, which have remained in the Bauman University collection until now. Among the models of the collections were those developed by Orlov and constructed in IMTS workrooms. Unfortunately, these models cannot be found.

2.2. Chebyshev's mechanisms

Pafnuty L'vovich Chebyshev was born in 1821 in the village of Okatovo in the Kaluga province. His primary education at home from the family. In 1837 he entered the faculty of philosophy of Moscow University, from where he graduated in 1841. In 1843 he passed examination for a master's degree of mathematics, and in 1846 he defended a thesis on "Experiment of the elemental analysis of probability theory". In 1847 he moved to St. Petersburg and went to work in a military academy at St. Petersburg. In 1849 Chebyshev defended a thesis for the doctor's degree of mathematics and astronomy on the theme "Theory of congruences" and two years later he was elected a extraordinary professor of Petersburg University. In 1853, Chebyshev was elected an adjunct professor, and ordinary academician of Petersburg Academy of Science of applied mathematics in 1859. He began teaching by lecturing on practical mechanics, the interest of which had an impact on his mathematical work. At different times Chebyshev taught courses of higher algebra, analytical geometry, number theory, single integral and probability theory. He created a famous mathematical school, which included outstanding mathematics A.M. Lyapunov, A.A. Markov, S.N. Bernshtein, etc. The theory of the best approximation of the functions by "Chebyshev polynomials" was born thanks to his passion for practical tasks. In 1852 in the Worldwide Exhibition in Paris he drew attention to the spotty wear of the rod of Watt's steam engine.

The article "About parallelograms" was where the first challenge of the parameter optimization of mechanisms was solved. Chebyshev died on November 26, 1894 from heart attack [44].

Linkages have been in to use in Europe since about the 12th century. During 12th–17th centuries their development progressed very slowly as the process of manufacturing joints was laborious. The establishment of precise prismatic pairs was complex, for example, for guides of steam engines cylinders. Therefore since the mid-18th century various constructions of linkages for the transformation of the linear motion to the rotary motion of a body were developed mainly empirically. The development of these mechanisms was not only the goal of the mechanism "for saving fuel and the strength of the machine", patent rights, specifically their circumvention played a substantial role too. There are well-known steam engines of Papin (1690), Savary (1698), Newcomen (1712), Leupold (1724). The first Russian steam engine was that of the Russian engineer Ivan Polzunov constructed in Altai (1766).

At the time when Chebyshev attended to this issue, the most sophisticated machines had been Watt's. To convert the rectilinear movement to rotary or to a swinging movement, Watt created several options of mechanisms, including empirically, by "trial and error method", two linkages (Figs. 2.4 and 2.5). One called reduced parallelogram was easier, but provided less accuracy of reproduction of linear motion, the second – full Watt's parallelogram, was more complicated, but also more accurate.

As Chebyshev wrote [46]: “Rules follow Watt, as a parallelogram device, and could serve as a guide for practice only until it met the need to change its shape. With the change of the form of mechanism new regulations were required”. With regard to the rules of Watt, French engineer and mathematician Bour wrote that they “were created for purely secondary benefits, while the matter was of principled importance: to an extent possible to reduce the deviation. Properly speaking, it was very difficult to start the issue, which explains why it remained intact until the work of Chebyshev” [50].

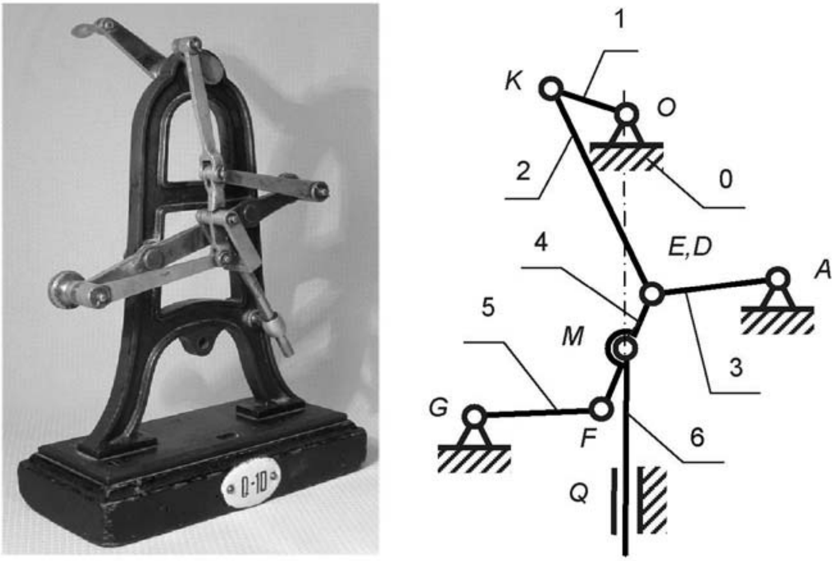


Fig. 2.4: Straight-line mechanism with reduced parallelogram Watt (Q-10)

Several works of Chebyshev were devoted to the research of parallelograms [45–49], one article written in 1869 deserves special attention [48]. In this work, Chebyshev used the series decomposition method and found a variant of assemblage of Watt’s parallelogram and the ratio of the lengths, its links providing minimal deviation linearity of the trajectory of the anchorage point of a piston at the desired site. However, we must say that in Chebyshev’s mechanism, in contrast to Watt’s parallelogram, only one point of the mechanism moves in a straight lines. This is due to the fact that in on effort to improve the accuracy of the linear motion, Chebyshev stepped back from the ratio of lengths of the links of the parallelogram mechanism (Fig. 2.6).
In addition, Chebyshev proposed his version of the straight-line mechanism for reproduction of linear motion, which is known as the “Approximate straight-line generating mechanism of Chebyshev”. In the Bauman University collection there are two models of this mechanism. The first is ligneous (Fig. 2.7, right) and was produced in the IMTS workrooms. Second (Fig. 2.7, left) is part of a collection of Reuleaux’s mechanisms and was produced in Berlin in the workrooms of G. Voigt. In the collection of mechanisms of the Department of TMM, there are several models with Chebyshev and Watt’s parallelograms.

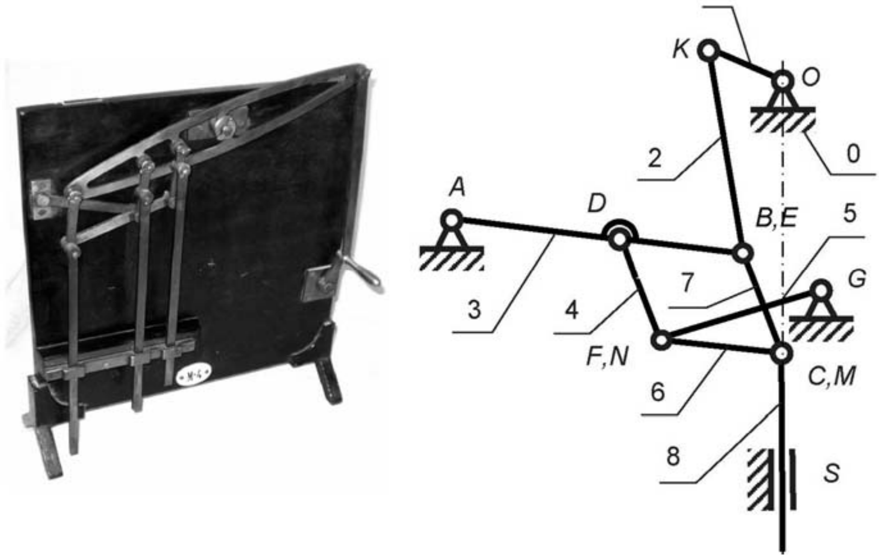


Fig. 2.5: Straight-line mechanism with a full Watt's parallelogram (M-4)

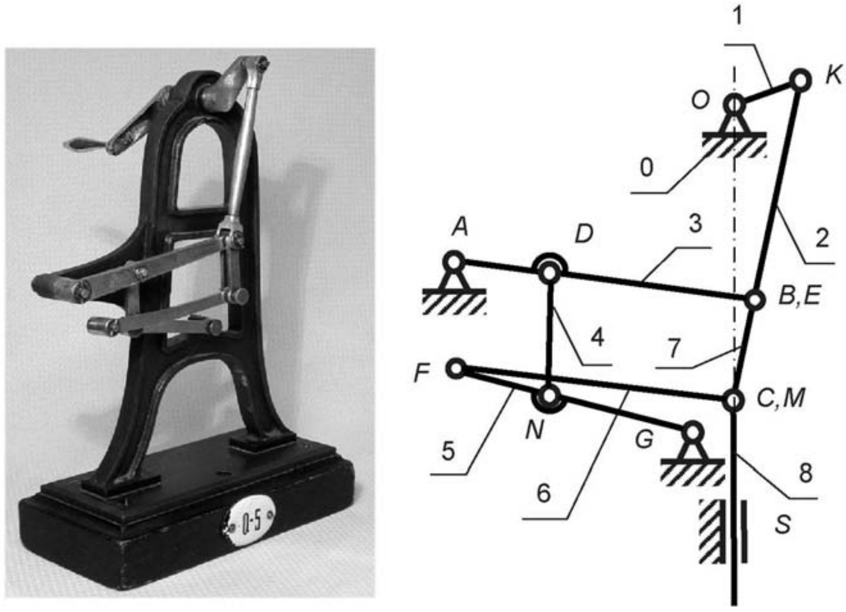


Fig. 2.6: Chebyshev's straight-line mechanism (Q-5)

Fig. 2.7: Models of "Chebyshev's priamilo" – straight-line mechanism

In his article [49], Chebyshev found ways to reach the largest, and the desired approximations to linear motion and, in fact, proposed a new, original system for a steam engine. This machine, in the view of some Russian naval engineers, could be better and more economical than the steam engines of Penn and Cave, because it would combine movement in a straight line with a continuous rotary motion; a part of the parallelogram replaced a con-rod, thus transmission of the piston's movement to the shaft was immediately achieved. Chebyshev's steam engine was single-cylinder vertical. The stock of the piston of the cylinder joined with the shaft. A hand wheel was placed there. It reduced unevenness of rotation. Rotation of the shaft was provided by an articulate parallelogram. Chebyshev's steam engine, its model and kinematic scheme of mechanism are shown in Fig. 2.8.

Naval engineers appreciated the quality of Chebyshev's steam engine. Indeed, one of them asserted that "it is enough to look at the five main systems (of steam engines), to be sure how much the steam engine with Chebyshev's mechanism can complete with every of them in its simplicity" [44]. Among other machines of the same kind, Chebyshev's steam engine, in his view, had the following advantages:

1. The easiest way of transmission is more profitable, in some respects, than in trunk cars and in cars with tilting cylinders because, without a trunk and tilting cylinder, we lead steam to the cylinder using common suitable solutions. We do not limit it to narrow corridors (as in a tilting cylinder), and cool it (such as in a car with a trunk).
2. Cost-saving of steam is possible because the system admits to leading superfine ways to cut steam, without mach effort, as in the machine with a tilting cylinder.
3. Manufacturing a cylinder for the machine should be cheaper than a trunk cylinder and tilting.
4. Changes (deformation) of the hull in the car cannot have significant influence on it because the piston rod is not led by a slide, and is directed by a parallelogram.
5. Compactness of the machine when its parts are well-located, should be the same as

in tilting cylinders and more profitable than the machines of all other systems, thus the machine of such a system should be very beneficial to the vessel.

6. It is hoped that the machine with Chebyshev's mechanism will be very cheap".

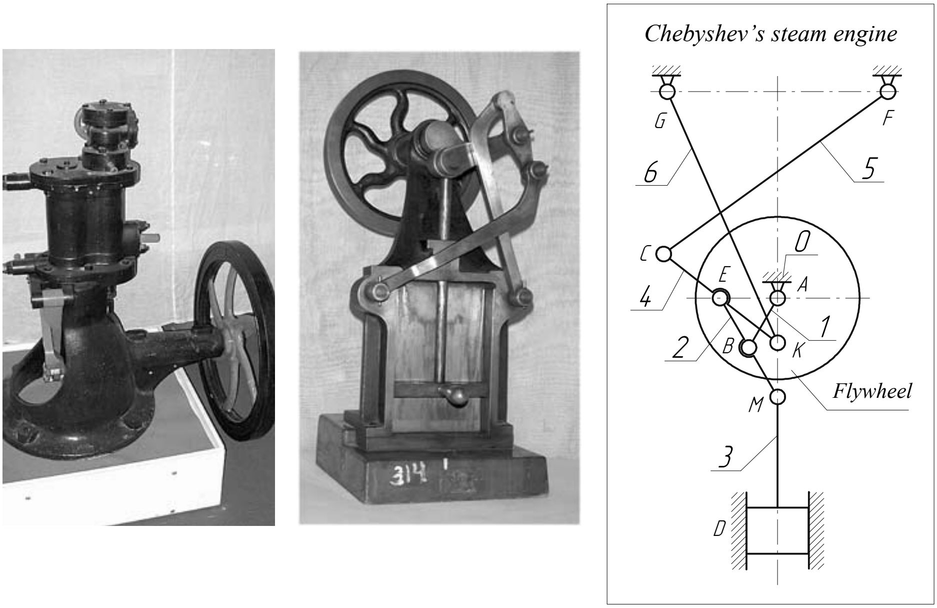


Fig. 2.8: Chebyshev's steam engine, its model and the structure chart of the mechanism

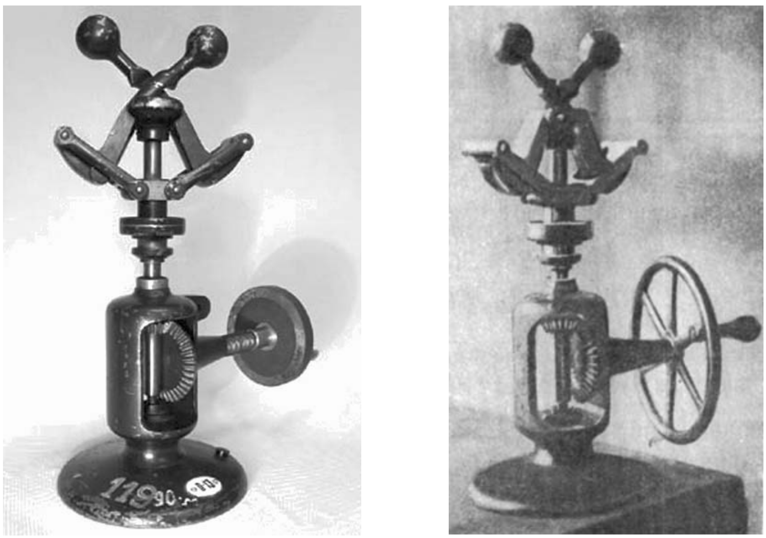


Fig. 2.9: Cebyshev's inertia governor

Chebyshev's steam engine was exhibited at the World Fair in Philadelphia in 1876. It was established and tested in the High Technical School in Moscow. This test showed that the use of Chebyshev's parallelogram is possible, though it turned out that further improvements necessary to achieve quiet and stable running. At present it is kept in the museum of Bauman University.
One of the six models of centrifugal regulators stored in the collection of the Cabinet of Mechanisms of the TMM Department, was calculated and designed by Chebyshev. In Fig. 2.9 the left photo shows this model; right – photo of a similar model from the article about mechanisms, established by Chebyshev [51]. One distinctive feature of the regulator is a complex configuration of weighted levers. This design provides a significant moment of inertia of goods and their balance: the static moment of the top of the lever become balanced by the moment of its low part. In this design inertial force doesn't overcome the weight of the goods and the sensitivity of the regulator to change the angular velocity is higher than in other construction regulators.
In total, Chebyshev developed over 80 mechanisms and devices, among them the plantigrade machine, gravity chair, rowing mechanism, inertia governor or centrifugal regulator, calculating machine. Models of these mechanisms are stored in the Polytechnic Museum, BMSTU, Moscow State University, University of St. Petersburg and also in several foreign museums.

2.3. Demonstration models of N. Joukovsky

Professor N. Joukovsky was not only an outstanding mathematician and technician but also a remarkable engineer. It has played an essential role in the history of the "Applied Mechanics" Department. He was on friendly terms with Professor F. Orlov and highly estimated his course on applied mechanics [11]. In 1895 he invited the young engineer N. Mertsalov to the post of senior lecturer and curator of the applied mechanics's cabinet. He was the teacher of Professor L. Smirnov. The high level of the courses "Analytical Mechanics" [10] and "Theoretical Mechanics" [9] read by him allowed an increase in the theoretical level of applied mechanics.
Undoubtedly, the basis of the works of N. Joukovsky have been connected with problems of aeromechanics, aeronautics and hydromechanics. However, some of his works are devoted to problems of applied mechanics, including the theory of regulation, dynamics of a kinematic circuit with one degree of freedom [52]. The course "The applied mechanics" which he read in the Moscow Practical Academy of Commercial Sciences, published by lithographic in 1901, was kept by a student of IMTS, V. Vladimirov [53].
It is known that in the mechanics course N. Joukovsky widely used demonstration models. However, at present we have only three models. These are a gyroscope on a gimbals suspension (Fig. 2.10), an example of the proof of the existence of non-sliding arches (Fig. 2.11), and a device for the proof of sliding of an extensible belt (Fig. 2.12).

Fig. 2.10: Model of a gyroscope on a gimbal suspension

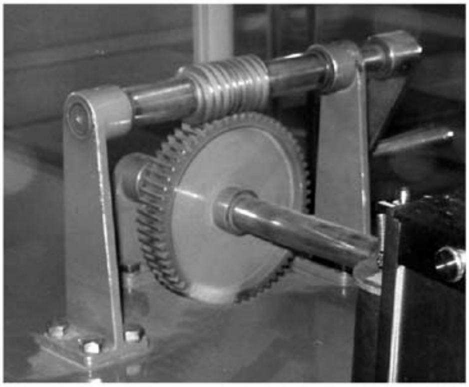


Fig. 2.11: Model for the proof of existence of non-sliding arches

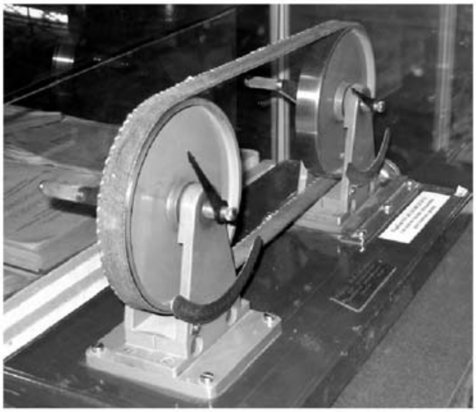


Fig. 2.12: The device for the proof of sliding of an extensible belt

2.4. Further accumulation of the collection by Professor L. Smirnov

From 1898 to 1929, the Applied Mechanics Department was headed by N. Mertsalov. We do not have any information concerning Mertsalov's contribution to the development of the collection of mechanisms. There are photographs of two Reuleaux-Voigt models and a wooden Chebyshev straight-line mechanism model in his book [17]. It is mentioned that the last one was manufactured in the department. Mechanisms of rotary machines are described in detail in the book. Engagements used in these mechanisms are illustrated. It can be assumed that when Mertsalov's book was published there were no Schröder models (Fig. 2.13), which were intended for demonstration of these engagements, in the collection yet. They were evidently purchased in Germany either when Mertsalov headed the department, or later.

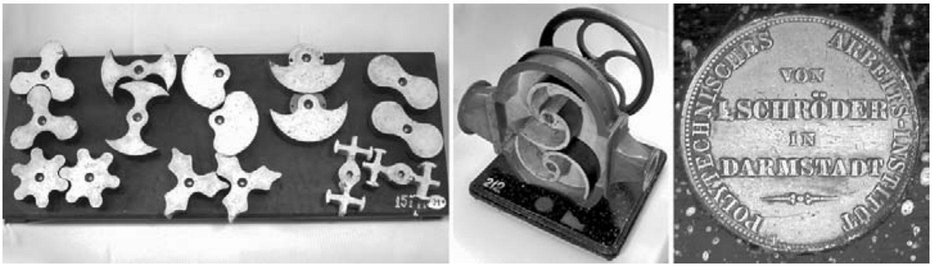


Fig. 2.13: Schröder's model for demonstration of the gearings used in a rotatory blower

Mertsalov had not only extensive theoretical knowledge but also practical skills in operating mechanisms and machines because when he was young he was a mechanical engineer at factories in Russia and Germany. He paid much attention to the collections of mechanisms of the Applied Mechanics laboratories of Moscow University and IMTS–MHTS. At that time, many models were developed at the TMM department and then manufactured in the School's workshops.

Since 1929 the department was headed by L. Smirnov (1929–1948). Under his direction, the collection of mechanisms continued to be enriched with new models, the majority of the models being developed by members of the department. The most significant contribution was made by L. Reshetov. On Smirnov's initiative, he was engaged in designing and making of gearings and cam mechanisms and increased the collection with new items (see Chapter 3).

Some mechanisms were designed by Smirnov in person. In our opinion, the "harmonizer" is the most interesting of them. The device was meant for synthesis of a function by given values of Fourier series coefficients. The idea of its creation apparently resulted from Smirnov's work concerning steam engine mechanics. Indeed, the moment of forces acting on a piston reduced to the shaft of a crank can be represented by the function shown in Fig. 2.14. The large extent of the nonzero load section is its characteristic feature. It is proved that such a function can be approximated by six Fourier series harmonics with adequate accuracy Fig. 2.14 (curve 1) – by three

terms with odd and three terms with even coefficients of the series. Amplitudes are parameters of these harmonics. They can be easily computed with the help of certain approximate formulas. Undoubtedly, to check whether the synthesized function converges to the given function two more harmonics were to be calculated (an approximate formula allows the computing of 12 terms of Fourier series – Fig. 2.14, curve 2).

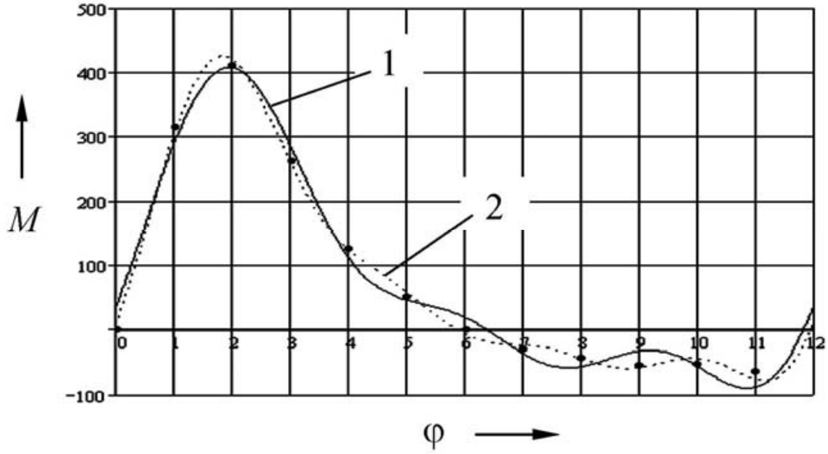


Fig. 2.14: Example of a reduced moment approximation by Fourier's row

Thus, the scientific interests of L. Smirnov agreed with important trends of the development of mechanics during the first half of the 20th century – with the development of mechanical calculators and mathematical equipment and tools (planimeters, calculators, analyzers, etc.) The harmonizer (Figs. 2.15, 2.16) designed by L. Smirnov and manufactured in BMSTU in 1934–1945 also belongs to such mechanisms. Two copies of the mechanism have survived: the first one is kept at the BMSTU museum, the other one, at the TMM department mechanism laboratory.

There are various designs of devices meant for synthesis of a function by its given values. Some of them are considered in [53]. Smirnov's "harmonizer" is intended for the approximate synthesis of a given periodic harmonic function by eight values of amplitudes of corresponding harmonic series terms computed by applying approximate formulas. The result is a complex polyharmonic curve that approximates the given function.

The "harmonizer" consists of eight controlled amplitude sine-mechanisms. On the left part of Fig. 2.15 one can see four mechanisms that describe various amplitude sine-curves. Correspondingly, on the right there are four mechanisms that describe various amplitude cosine-curves. Output sliders of each of the mechanisms through a system of blocks, are sequentially connected to a writing device slider by a rope. A cylinder with a sheet of paper on which the resulting curve is traced is connected with the input shaft of the crank-rocker mechanisms via a worm gearing. The power shaft is

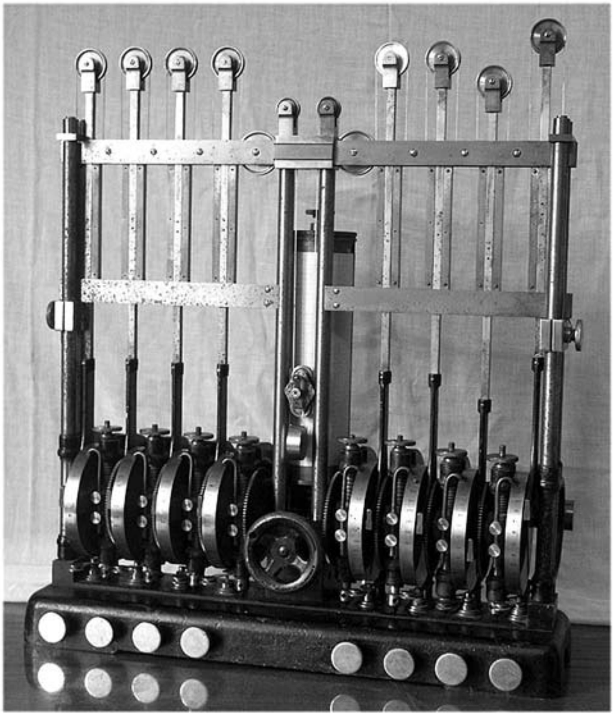


Fig. 2.15: The harmonizer designed by Smirnov

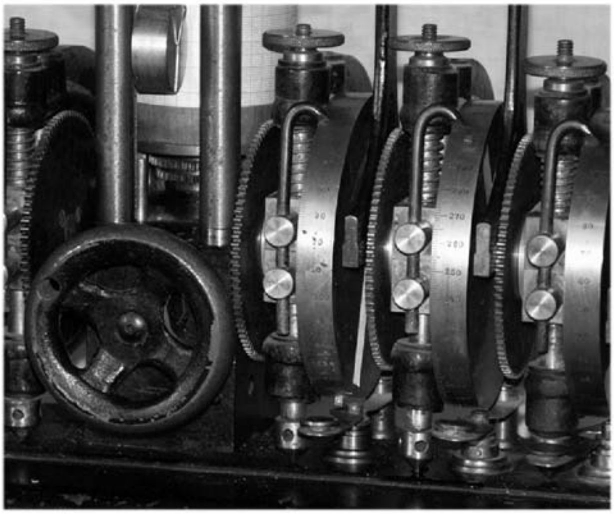


Fig. 2.16: The elements of the harmonizer's drivers

connected to the input shaft through gearings with gear ratios U = 1,2,3 and 4. There are two sine-mechanisms for each gear ratio: one of them provides an output sine curve, the other one – output cosine curve. Cranks of the sine-mechanisms are connected to the wheels of the gearings by screw mechanisms that permit to varying sine-curve amplitudes. After setting the amplitudes of the harmonics being summed, the shaft of the mechanism rotates and a pencil traces a sum curve on the coordinate paper sheet fixed on to the block. It takes no more than 20–30 minutes to set up the device and trace the curve.

2.5. Professor Leonid Reshetov's contribution

The major part of the collection of mechanisms is connected with the name of Leonid Reshetov. In 1930 after graduation from MMMI, Reshetov remained at the TMM department where he worked as an engineer and was simultaneously engaged in teaching activity. Because of his work he was closely connected with industry, especially with railway transportation. One of his first works was to research of theory of corrected evolvent cog-wheels. Reshetov made a big contribution to designing cam mechanisms. The results of this work were reflected in the monograph [29]. Reshetov was the a Honored Inventor of Russia, the author of more than 50 inventions. The majority were realized by industry. The big collection of kinematic pairs and statically definable connections, models of mechanisms of controllers, planetary mechanisms, hour calendars, cardans and many other devices were designed by him. In developing his ideas he almost always checked them on test mock-ups or models. All of them were created from his drawings and sketches. He personally made many models. He often made preliminary draft models using children's meccano sets.

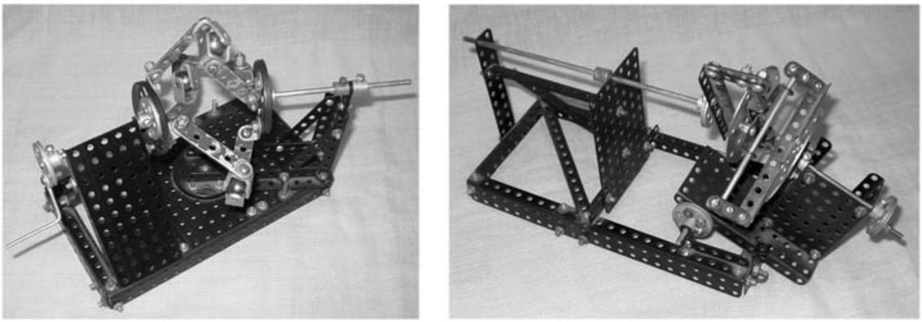


Fig. 2.17: The models of universal joints collected by L. Reshetov from the meccano sets

Examples of such models of universal joints are shown in Fig. 2.17. During all his time in the TMM department, Professor L. Reshetov was the curator of the collection of mechanisms. In 1974 Leonid Reshetov tried to create albums with descriptions of models. Only one album was completed. Work with the others, unfortunately, wasn't completed. In 1960–1970 the TMM course was considerably reduced. The number of employees of the department was also reduced, decreasing the number of premises used

by department. The collection of mechanisms had significant losses: many models were transferred to the Polytechnical museum, museum of BMSTU, and to departments of other institutes. Some of the models were replaced and left without supervision; many models were lost or destroyed. This trouble concerned not only BMSTU, but also many other organizations. In particular, the models of the Moscow University, Moscow Power University, Moscow Aviation University were lost. It was doubtless to the merit of L. Reshetov that the MSTU TMM department kept a considerable part of the collection of mechanisms.

2.6. Contribution of Professor Vladimir Gavrilenko's scientific school

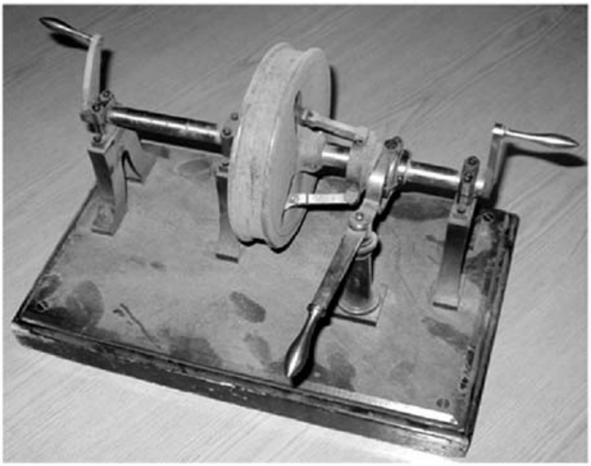
Vladimir Gavrilenko, son of the first elective rector of IMTS A. Gavrilenko, headed the TMM department in 1962. This was the time when TMM was reduced, and its place was taken by new disciplines, basically connected with electronics and computer facilities. During this uneasy period, Gavrilenko managed to keep the basis of the collective of the department, to prepare a base for a course revival. As was already remarked, an incomplete a collection of models was kept. Revival of the department began in 1968. At this time new, young employees who now form basis of its collective were sent on faculty. During these years the TMM course was essentially modernized. Students began to perform two course works or two house tasks. The academic year project became more composite. New laboratory equipment ware was bought and new laboratory works were created. At this time, complete sets of models of the mechanisms, developed by SouzVuzPribor (special design offices which provide the educational process of universities of the USSR with models and devices) were bought. A scientific group under the direction of Gavrilenko whose problem was carrying out research and developmental works on crank-planetary and wave-tooth gearings, was organized in 1966 in the TMM department. On the basis of the theory of involute gearing, developed by Gavrilenko, special gear drives with internal involute gearing and wave-tooth gearings were projected in this group. Results of the work of this group found their place in a collection of mechanisms. Another source of updating of the collection during those years were the models created by the post-graduate students of the faculty resulting from scientific researches.

2.7. New times' problems and perspective views

Since 1978 and until now, BMSTU's TMM department has been headed by Academician of the Russian Academy of Sciences Professor Konstantin Frolov. For last decades the collection of mechanisms hasn't undergone many changes. There were neither significant receipts nor essential losses. In 1979 the educational laboratory and department have exchanged places, the laboratory has moved. Now the basic part of the collection of mechanisms is placed in the educational laboratory.

In 2004 the curator of the collection of mechanisms, Valentin Tarabarin, was appointed one of authors of this book. Under his management, ordering of the collection began, with the photographing and video shooting of models, and descriptions made. The historical part of the collection was allocated in separate section. Part of the missing models were found. In total, it has recorded more than 30 models, many of which concern rare models of 19th century. In the photo shown in Fig. 2.18, are shown some models after their extraction from base. In 2004–2005 contacts with the Sibley School

of Mechanics of Cornell University were established. This university possesses the largest collection of models of Reuleaux – Voigt and also a considerable historical library on applied mechanics. The curator of this collection Professor Francis Moon, heads the section on assemblies of models of mechanisms at the commission IFTOMM on TMM history. An Internet website devoted to the collections of Reuleaux mechanisms is located to the address: kmoddl.library.cornell.edu. On this site, a number of models from our collection is presented. In 2005 a symposium devoted to the 175th anniversary of BMSTU held a workshop on TMM history. More than 20 scientists from 15 countries of the world took part. Participants of the meeting were interested in the BMSTU collection of models of mechanisms and estimated it as a monument to the history of science and techniques on a world level.



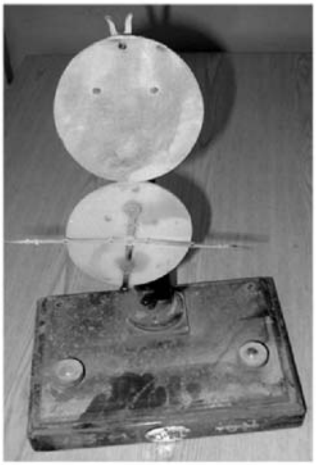


Fig. 2.18: The models of Reuleaux-Voigt taken from subfield

Now the collection numbers more than 600 models. Models are in various conditions. Most models require restoration and the carrying out of procedural work: cleaning, lubrication and painting. More complex problems emerge with models of 19th century because qualified restoration of many models is necessary. There are problems with the establishment of time and place of manufacture of some models. For tabulation of the electronic catalogue, it is necessary to describe and professionally photograph models, to lead to a qualitative video shooting of them. In the educational laboratory there is not enough room for the organization of a modern museum exposition, and there are no special cases for displaying exhibits. We hope that all these difficulties will be overcome and the collection of mechanisms will be presented with dignity in BMSTU.

2.8. Classification of mechanisms of the Bauman University collection

There was no comprehensive inventory of the collection of models made over the lifetime of the TMM department (that is, over 150 years). In the 1970s the arrangement of books of models began under the direction of Professor L. Reshetov. The books contained photographs of models and their brief descriptions. Teaching staff and postgraduates of the department were involved in the work. Only one of the books was

completed (Fig. 2.19). Photographs, partially completed schemes of mechanisms and descriptions were prepared for the other books, but the work remained unfinished.

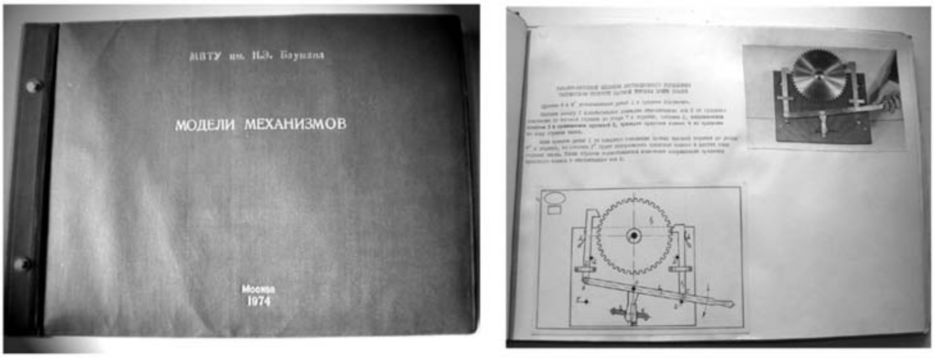


Fig. 2.19: An album with descriptions of models from the BMSTU collection and one of sheets of the album with a description of the mechanism of a regulator

Nowadays the majority of models of the collection have an enamelled plate with the model's symbolic notation engraved on it. The notation consists of a Roman alphabet letter and a number. Analysis of models of the collection allowed us to disclose the following correspondence between notation letter and mechanism type: *A* – kinematic pairs; *B* – linkages and gear-and-link mechanisms; *C* – mechanisms of cardan joints (universal joints); *D* – screw mechanisms and wedge mechanisms; *E* – cam mechanisms; *F* – friction mechanisms (clutches and couplings, braking devices, belt transmissions); *G* – non-round wheel gearings, counter mechanisms, reverse trundle transmissions; *H* – planetary trains and differentials; *I* – models of connecting profiles; *J* – toothings; *K* – screw, worm and spiroid gearings; *L* – intersecting axes gearings (bevel gearings); *M* – planar linkages; *N* – mechanisms of planetary trains and differentials; *O* – inertia governors; *P* – mechanisms of devices with pneumatic actuators; *Q* – straight-line-and-guide linkages; *R* – steam engine mechanisms, crankless mechanisms, mechanisms of hydraulic motors; *S* – simple gear mechanisms; *Z* – gear clutches. The authors did not succeeded in finding ground or description of the mechanism classification system in the available literature. Inquiry of senior members of the teaching staff of the department did not permit us to determine the authors of the classification either. Our research allowed us to ascertain the following. On the left of Fig. 2.20 photographs of three models of the collection from Mertsalov's book [17], which was published in 1916, are shown. Here one can see that models of Leonardo da Vinci's machine and two models of Chebyshev's straight-line mechanism have plates with engraved numbers "111", "3" and "22" fixed on them. Thus, there was a simple notation without any division into types used when the photographs of these models were taken.

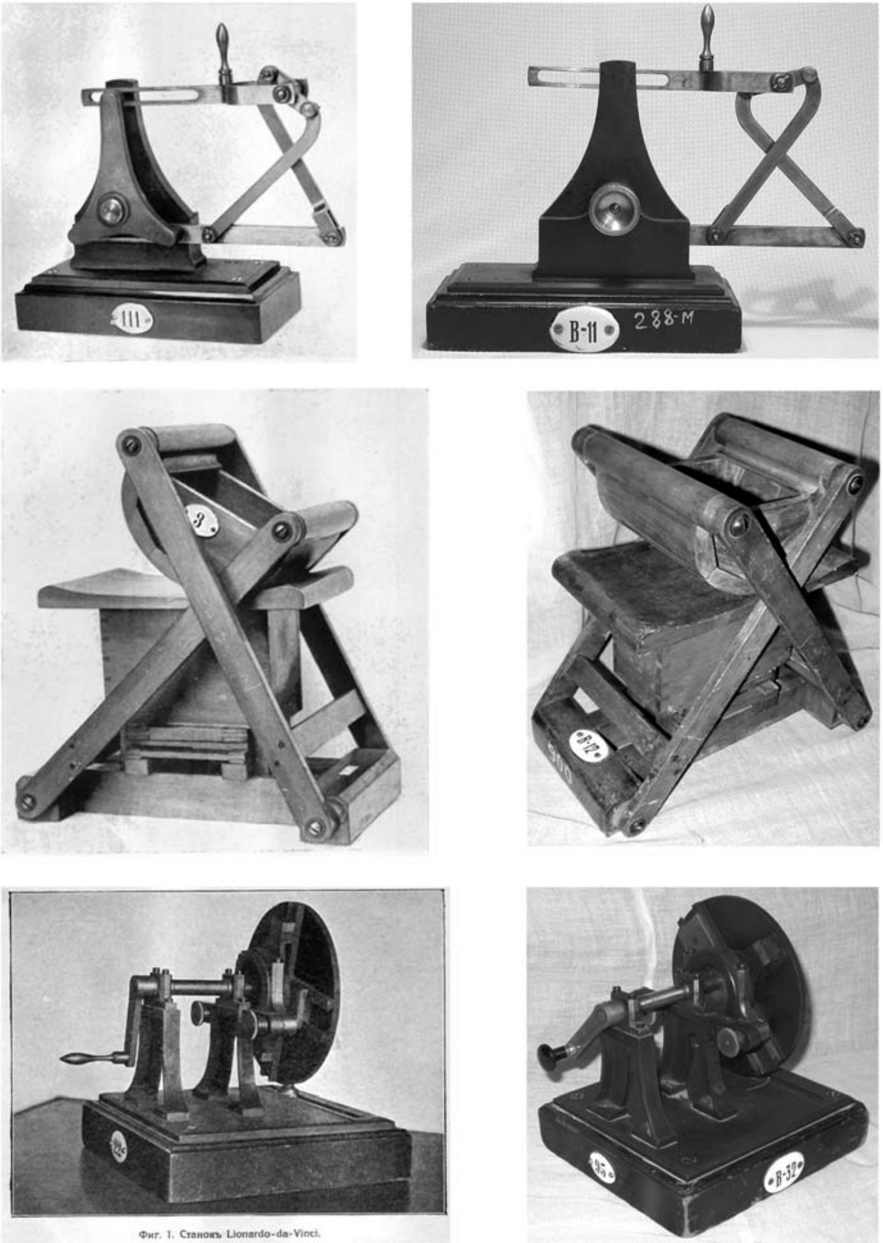


Fig. 2.20: Left, photos of models from Mertsalov’s book [15], and right, modern photos of the same models

On the right of Fig. 2.20 present-day photographs of the same models are shown. The models have plates with symbolic notations “B-12”, “B-11” and “B-32” there, the model of Leonardo da Vinci’s machine having two such plates: “95” and “B-32”. Therefore, between 1916 and 1960 the numeration of models was changed twice. First, Leonardo’s model had plate “22” replaced by plate “95”, then plate “B-32” was added. N. Mertsalov was head of the TMM department till 1929. Then the post was taken up by L. Smirnov. Engineer L. Reshetov began working at the department after graduating from the School (that time it was called MMMI) in 1930. In 1937 he left the department. When he returned in 1951 he became its head and headed the department till 1962. Since then and until his retirement he supervised the collection of mechanisms and perhaps he was the one who introduced the alpha-numerical classification of mechanism models. The fact that models developed by Reshetov in the 1930–1970s have the same symbolic notation also counts in favor of the fact. Section “B” appears to be the largest one. It includes various mechanisms: planar linkages, gear-and-link mechanisms, trundle clutches, straight-line-and-guide linkages, Leonardo’s machine. Sections “H” and “N” consist of types of mechanisms close to the those in section “B”. Thus, sometimes it is a problem to define what section the mechanism belongs to. For example, it is not understood why all typical planetary trains are attributed to section “H” and Lahire’s mechanism, to section “N. Similar problems occur with other mechanisms, too. Many mechanisms of the collection do not have classification numbers or inscriptions, part of the models have them damaged (unreadable) or do not have them at all. Therefore, much remains to be done concerning classification and systematization of models of the collection. Unfortunately, all known mechanism classification systems suffer from the same disadvantage.

3. RUSSIAN MODELS IN THE BMSTU MECHANISMS COLLECTION

3.1. Models of kinematic pairs and statically determinate connections

A part of the models of kinematic pairs and connections created by L. Reshetov occupies a considerable place in the collection (Fig. 3.1). The total number of models in this part is more than 50. On can find a description of most of the pairs in L. Reshetov's book "Design of "Rational" mechanisms" [31]. In this book a combined table where the author shows famous types of kinematic pairs is presented.

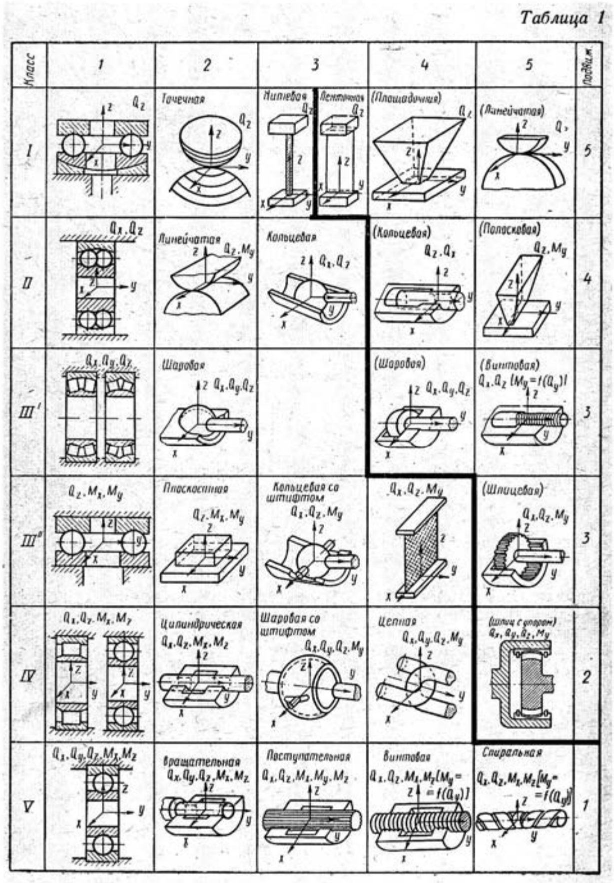


Fig. 3.1: The table of kinematics pairs in L. Reshetov's book

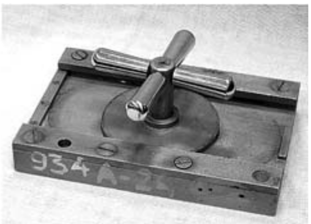
There is no possibility and no point in viewing all the models in our short review. Models of this part can be divided on four groups:

A. Golovin and V. Tarabarin, *Russian Models from the Mechanisms Collection of Bauman University.*

1. Models of kinematic pairs with hard links
2. Models of kinematic connections equivalent to these kinematic pairs
3. Models of pairs with soft constraints
4. Models of split motionless link connections

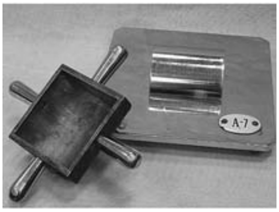
Models formed by two hard links (moveable and immovable) belong to the first group Models of translating, turning, screw, cylindrical, flat, spherical, higher and trajectory kinematic pairs also belong to first group models. Photos of 12 models of kinematic pairs presented in the Table 3.1.

Table 3.1. Models pf kinematic pairs with rigid links

 *Prismatic (splined)*	 *Prismatic (with force-closure)*	 *Prismatic (with form-closure)*
 *Spherical circular with a dowel*	 *Revolute*	 *Cylindrical horizontal*
 *Cylindrical vertical*	 *Plane (flat)*	 *Spherical circular*

Models of kinematic connections belong to the second group. A kinematic connection is a successive or parallel connection of several kinematic pairs for increasing relational movability between pair links (one – input and other – output) or for replacing sliding friction between elements of pairs on rolling friction.

Table 3.1. (continuation) Models of kinematics pairs with rigid links

 *Skrew vertical*	 *Trajectory*	 *Spherical*
 *Dotty*	 *Linear*	 *Rectangular*
 *Linear*	 *Strip line*	 *Chain*

For instance, a rolling bearing represents a revolute kinematic pair with balls and rollers between active surfaces of links which replaces sliding friction instead with rolling friction. In that case, mobility of balls and rollers is local. There are several models of similar mechanisms in the collection. Photos of this mechanisms are in Table 3.2.
Mechanisms where links are connected with soft constraints are widely used. Ropes, cables, belts and slivers of lift-transfer mechanisms and other machines belong to them. Kinematic pairs of this type are presented by two models in the collection of mechanisms. These models are shown in the photos in Table 3.3.

Table 3.2. Models of kinematics connections

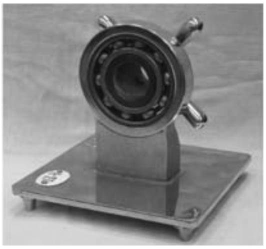
 *Revolvute kinematics pair*	 *Cylindrical kinematics pair*	 *Axial bearing*
 *Spherical axial bearing*	 *Spherical kinematics pair*	 *Roller bearing (cylindrical kinematics pair)*

Table 3.3. Models of kinematics pairs with soft constraints

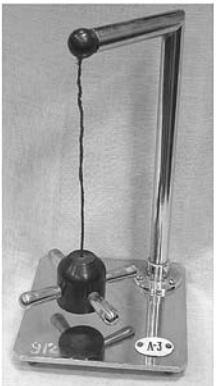
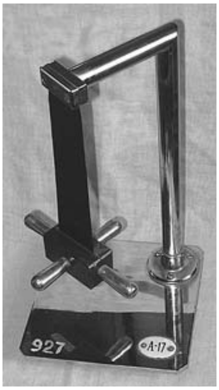
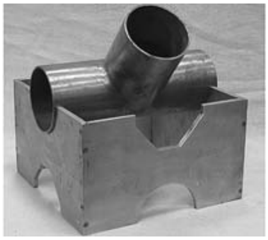
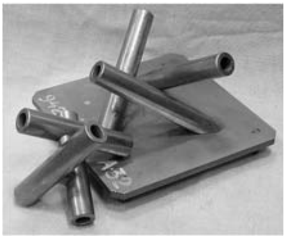
 *Filamentary-type*	 *Belt-type*

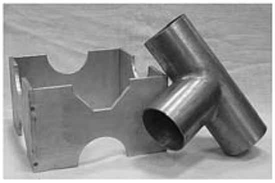
Table 3.4. Models of rational immovable connections

Assembled connections



Disassembled connection.





Connection of six first class kinematics pairs

Connection of three second class kinematics pairs

Connection of three second class kinematics pairs (or of 2 second and of 2 first)

Structures of mechanisms where kinematic pairs include soft constraints are set out in Reshetov's article published in the jubilee (125 years) collection of the TMM department "Problems of theory of mechanisms and machines" [55]. Immovable connections find an application in mounting machine frames on a base, in combining mechanism details into one link and combining separate machines into a unit. Carrying out the contact surfaces in accordance with the requirements of static definability, i.e. without redundant constraints, considerably reduces the laboriousness of mounting and assembling, reduces deformations of details and intensity of stress. Subsidence of the base or resizing link over temperature drop will not cause extra stress.
A statically determinate or rational immovable connection must put six constraints on the relative motion of links: three on the linear coordinate and three on the angle coordinate. These six conditions of constraint can be provided with different kinematic pairs, the number of them depending on a combination of classes and can vary from two to six.
The possibility of making only 30 variants of rational immovable connections was marked in the work: "… four from six pairs, four from five, seven from four, eleven from three and four from two kinematic pairs " [31].

Table 3.4. (continuation) Models of rational immovable connections

Assembled connections

Disassembled connection.

Connection of four first class and one second class kinematics pairs

Connection of fifth class KP and first class KP

Connection of fifth class KP and second class KP

Ready-assembled connections

Connection of fourth class KP and second class KP

Connection of fifth class KP and bolts

Connection of six bolts and 12 fifth class KP

Connection of five bolts, 8 fifth class KP, one fourth class KP and second class

Connection of six bolts and 12 fifth class KP

More numbers of rational connections of two links could be created using a set of additional links and kinematic pairs. The mobility of additional links must be equal to the number of constraints in kinematic pairs, i.e. relative mobility of two connecting links must be equal to zero. In Table 3.4, photos of 11 models of rational immovable connections of two links are presented. Models in this table are presented in two conditions: assembled and disassembled. Such a way of presenting allows more details showing constructions of the elements of kinematic pairs using photos and allows for the finding out of place and form of the contact lines and surfaces. Structural analysis of the connections is made here, constraints and nobilities are also defined, the absent of redundant or passive constraints is shown, i.e. connections are statically determined.

3.2. Linkages

3.2.1. Planar linkages

We shall examine some of the models of linkages from the collection. We shall begin with the mechanism of an inversed ellipsograph. Photo and type diagram of this mechanism are shown in Fig. 3.2. The input link 1 is connected with framek 0 by revolute kinematic pair A, and output – by cylindrical pair K. An input link 1 has two

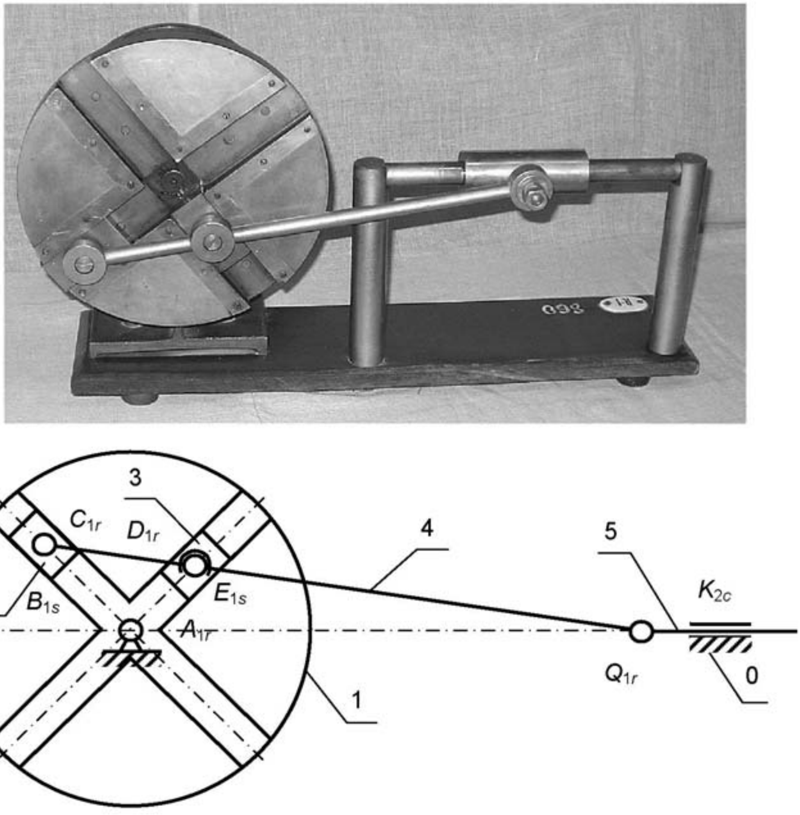


Fig. 3.2: Model of the mechanism of an inversed ellipsograph and type diagram of this mechanism

grooves perpendicular to each other. Sliders 2 and 3 are located in grooves. These sliders are connected to part 1 by prismatic pairs E and B, and revolute pairs C and D -by a coupler 4. Output link 5 is connected with coupler 4 by the revolute pair Q, and with a frame – by cylindrical pair K. There are five moving links, six one-moving pairs and one two-moving pair in the mechanism. If a link 1 is an input link, then the mechanism can be carried to second – class mechanisms with one three-dog group. There are five redundant constraints in the given type of diagram of the mechanism.

The mechanism is interesting because for one turn of an input link, the slider (output link) makes two cycles. The position function of the output link of the mechanism the result of which can be seen on the graphic chart, in Fig. 3.3. The model was designed in the TMM Department of BMSTU and manufactured in its workrooms.

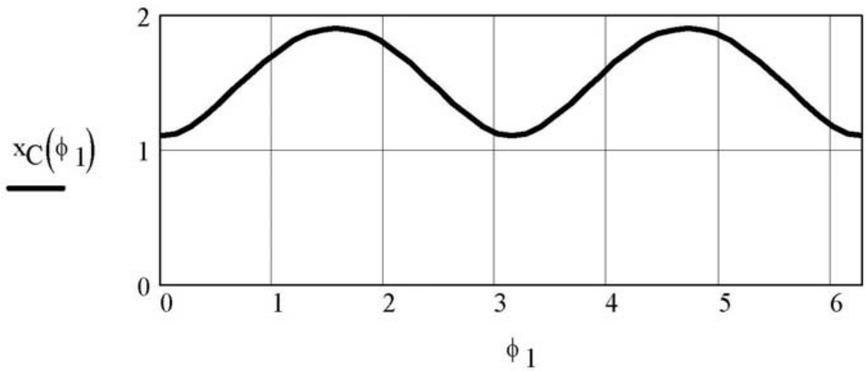


Fig. 3.3: Position function of slider of inversed ellipsograph

Further, we shall consider a model with a four-hinged claw mechanism (Fig. 3.4). The model shows the usage of coupler curve characteristics. For placing a claw mechanism's hook, the point of the coupler which makes an approximate rectilinear movement on the working part of a movement is used. The flywheel is structurally combined with the crank in this model. According to the accepted classification (n. 2.8) it is concerned to mechanisms of type "M". The model was designed and manufactured in the TMM department and intended for demonstrating of coupler curve properties during lectures on designing linkages.

Inversors concern theoretically accurate directing mechanisms. There is one such mechanism in a collection of models – the Peaucellier – Lipkin's mechanism. A version of this mechanism was invented in 1864 by French engineer Peaucellier. Four years later, in 1868, this invention of Peaucellier was independently repeated by Petersburg University's undergraduate L. Lipkin. With another version, Peaucellier just formulated a problem, the solution of which was given by Lipkin in 1868. Lipkin's results were published in 1871. Peaucellier himself found the solution only in 1873. In the same year, a similar solution was executed by other French mathematician R. Lemoine.

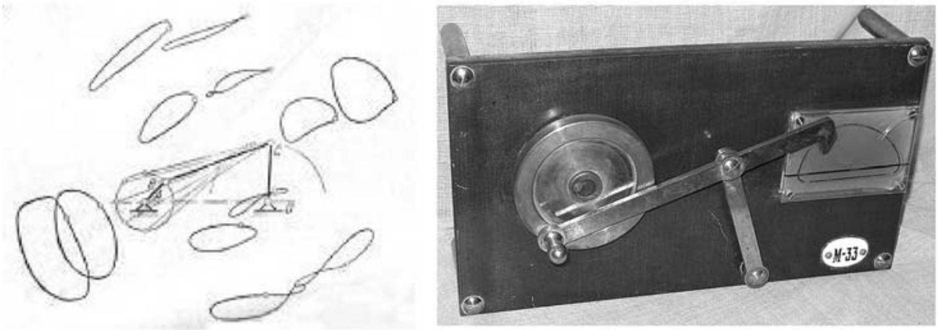


Fig. 3.4: Coupler curves of four-bar linkage and model of four-bar claw mechanism

A description of Peaucellier's mechanism is given in Mertsalov's book [17]. The mechanism "… consists of joint rhombus *BECD,* tops *E* and *D* of which are connected by links *DA* and *EA,* are equal between themselves, with a immovable point *O,* and the requirement *BO=AO* is observed. Under these conditions, the rhombus's top *C* will describe a straight line *HH,* perpendicular *to AO*".

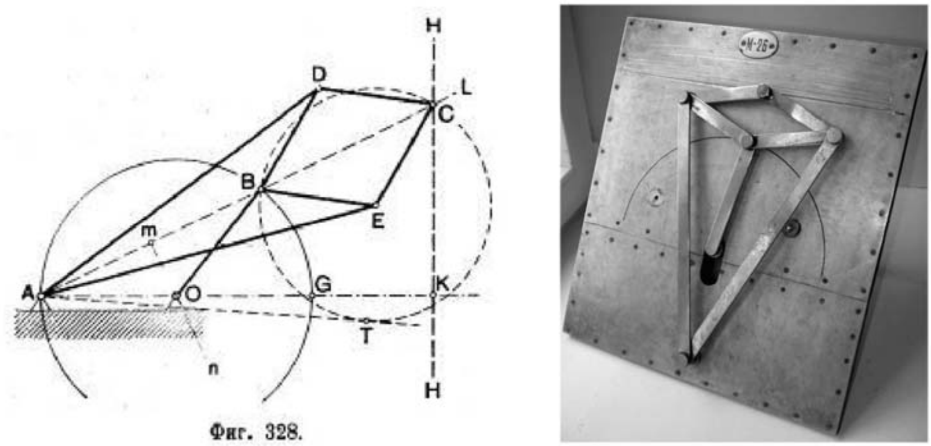


Fig. 3.5: Type diagram and photo of the Peaucellier-Lipkin 's mechanism model

The mechanism represented in Fig. 3.5 is Peaucellier's positive inversor as point *A* lays outside the rhombus *BECD* and the proportion is performed:

$$A - B \ = k^2 = AE^2 - EC^2$$

If radius *BO* is not equal to *AO,* point *C* will describe not a straight line but a circle. Mertsalov remarks, that Peaucellier-Lipkin mechanism – lacks a large number of joints. There are seven links and ten joints in the mechanism. Clearances in joints and their

wear will lead to significant deviations of trajectory *H-H* from a straight line. The model of the Peaucellier-Lipkin mechanism was designed in the TMM department of BMSTU and manufactured in its educational workrooms.
Under Pescar's offer, non-shaft generators of gases [31] began to be applied in diesel locomotives during 20th century. Pistons of a non-shaft generator of gases must move with identical speed. This is provided by linkage to the bar. Model P-2 illustrated in Fig. 3.6, represents such a leveling mechanism. The rational scheme of this mechanism from Reshetov's book [31] is shown next to this photo. Redundant constraints are eliminated in this scheme, an auxiliary slider is installed for this purpose and some revolute pairs are shown by spherical pairs. As well as the three previous models, this model was also manufactured by the TMM department.

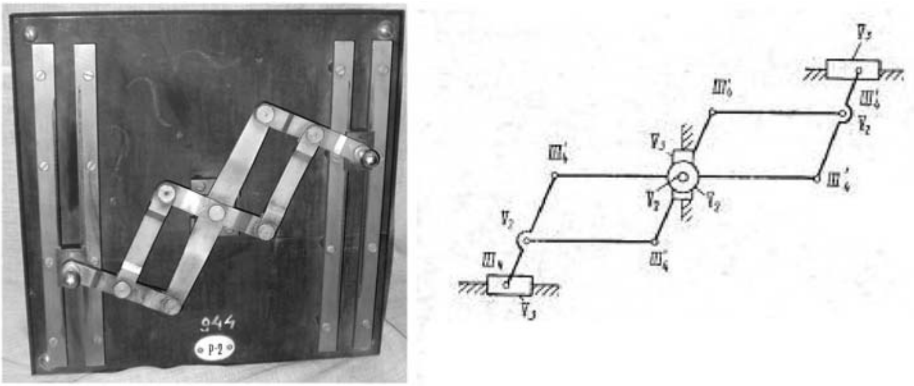


Fig. 3.6: The model of hinged leveling linkage of a shaftless gas generator and the scheme of redundant constraint elimination

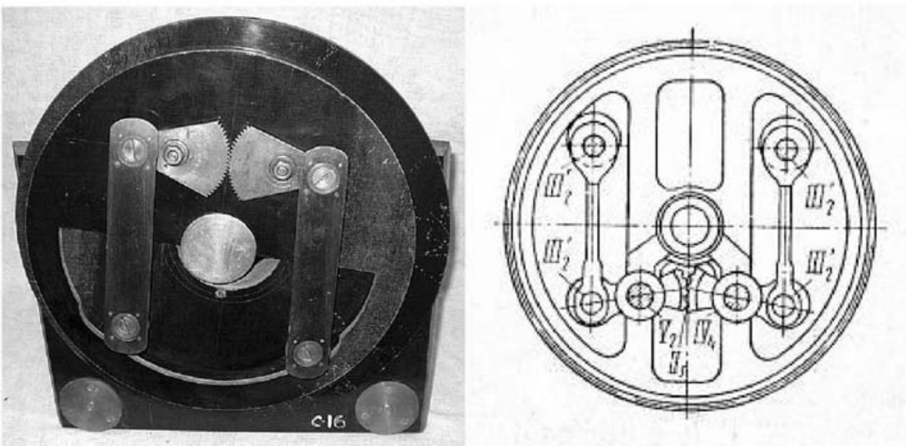


Fig. 3.7: Model and constructional scheme of the leveling mechanism of a locomotive drive

In passenger locomotives, the traction engine is attached to the carriage by a special linkage. Several designs of such mechanisms are known and some of the mechanisms are considered in Reshetov's work [31]. Mechanisms of this kind are necessary for reduction of the unsprung mass of the locomotive. They provide an opportunity for fluctuations the locomotive's case, in springs. Mechanisms should transfer rotary moment to the driving wheels of the locomotive and to leave the capability of moving to the remaining five coordinates: three linear and two angular. Model C-16 was executed under the scheme of Buhli's (Brown-Bovery) mechanism. It consists of two parallel levers connected by tooth gearing. The photo of model and the constructive scheme of the mechanism are represented in Fig. 3.7. The mechanism has six moving links. For elimination of redundant constraints, levers were established on the spherical pairs.

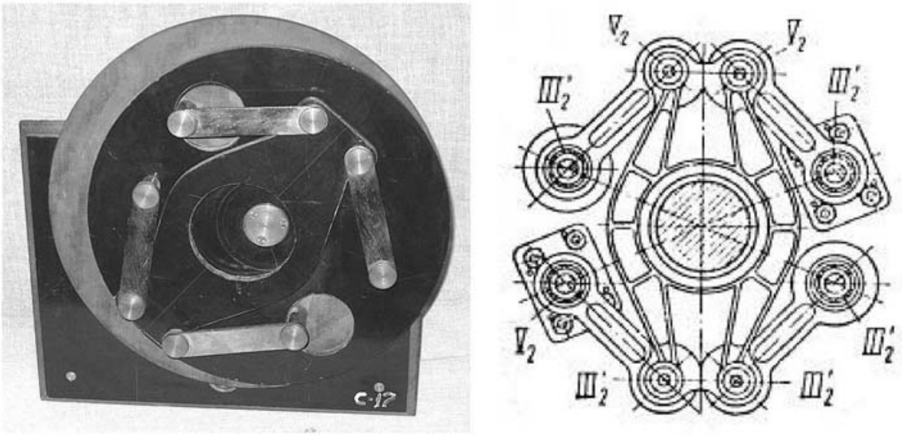


Fig. 3.8: Model and constructional scheme of leveling mechanism of Alstom 's locomotive drive

Model C-17 (Fig. 3.8) represents mechanism a similar which is named in [31] as the mechanism of Alstom's antiparallelogram. This mechanism has seven mobile links. The rotary moment is transferred by floating framework suspended on dogs. To hold the frame in the a set position, the number of dogs should be equal to three. Each of these dogs is installed on one revolute and on one spherical pair. It is necessary to establish a fourth dog on two spherical pairs to avoid redundant constraints. Each dog has a local mobility – rotation around their axis. For the reduction of forces in the directing device, it is necessary to increase the distance between dog supports. In Alstom's mechanism, all dogs work either by stretching, or by compression. The mechanism has small changeability of the transfer ratio which practically does not influence the working of the mechanism. Alstom's mechanism is not counterbalanced. At the displacement of axes, the floating framework moves on a circle the diameter of which is equal to the size of this displacement. For one turn of a wheel the center of mass of a framework bypasses the circle twice. Therefore on high-speed locomotives there are greater inertial loadings in these mechanisms. Models C-16 and C-17 were manufactured in the TMM department under L. Reshetov's initiative. One more model of locomotive mechanisms is the model of the fictitious pin mechanism. This mechanism serves to connect the

locomotive carriage when the place of the pin is occupied by the traction engine. Voigt's mechanism represents parallelogram joints, where the centers of the cross levers are connected to the case, and the centers of the longitudinal levers connected to the carriage (Fig. 3.9). This mechanism has two redundant constraints which create two tightnesses: at the assembly of the mechanism of case and at its connection with the carriage. It is necessary to cancel part of the levers and one hinge for elimination of these redundant constraints. These eliminated parts are represented by a dotted line in Fig. 3.9. It is necessary to make all pairs, except for one which remains revolute, spherical to completely remove the redundant constraints in the given mechanism.

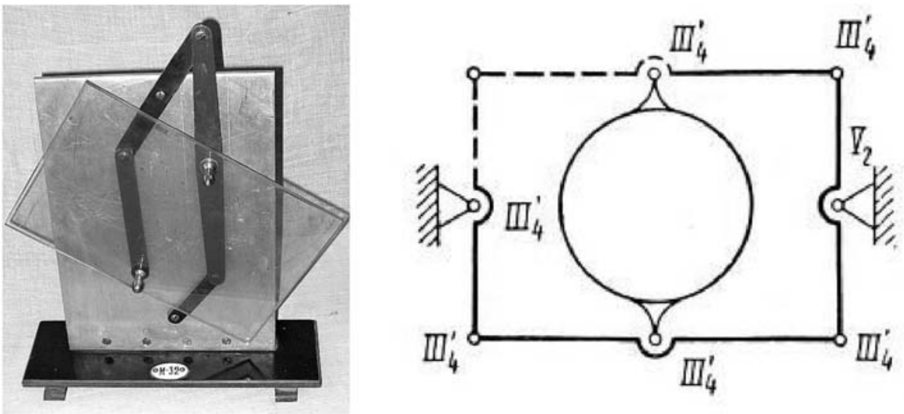


Fig. 3.9: The model and the scheme of the fictitious pin mechanism (Voigt's mechanism)

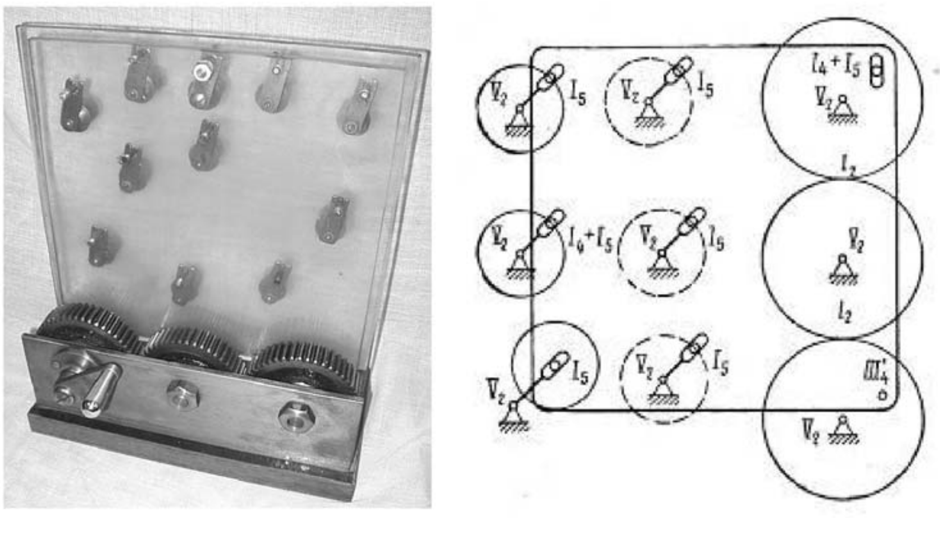


Fig. 3.10: Toothed linkage of a multispindle head

The model represented in Fig. 3.10, represents the mechanism of parallel cranks for the multispindle head of a drilling machine. Mechanisms of parallel cranks were not well received in drilling machines because of the large number of redundant constraints. Gear mechanisms with a greater number of intermediate wheels are usually used here. As the application of a parallel cranks mechanism essentially simplifies the design of multispindle head, L. Reshetov developed a design with a reduced number of redundant constraints. For then elimination of redundant constraints in the crank or in the coupler, the bush is replaced by a groove. In a dead (extreme) position, such a design eliminates redundant constraints. In other positions, mobility increases for 1. This mobility is eliminated by the introduction of additional constraints. In the given mechanism, tooth gearing with an intermediate wheel is used. The first wheel of this transfer forms a movement on a circle to a coupler through a crank, and the crank of the third wheel provides the additional connections, which eliminates the opportunity of the rotation of a rod (plate) in a plane of mechanism.

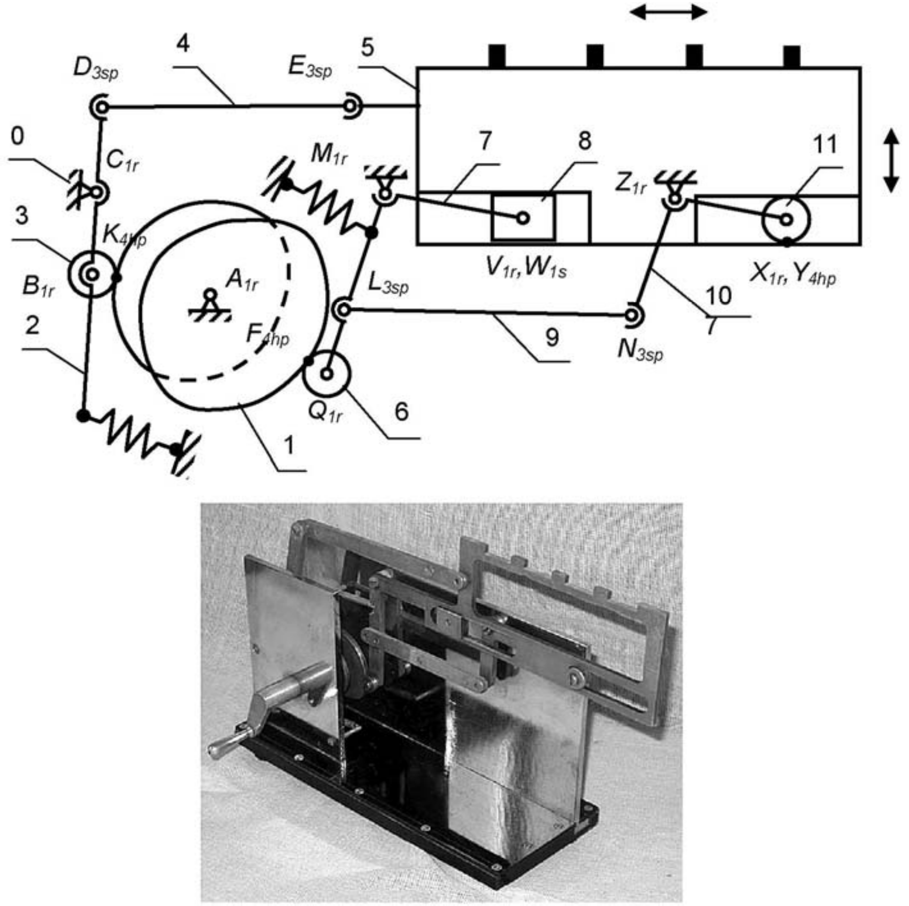
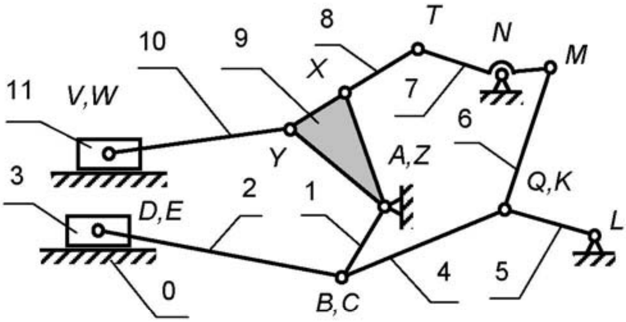
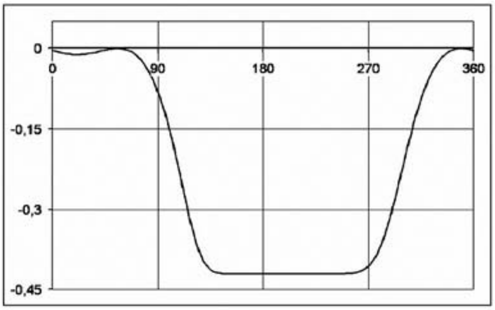


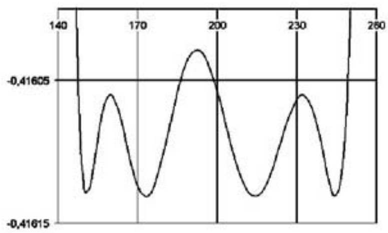
Fig. 3.11: The model and type diagram of the mechanism of a manual conveyor





The graphic chart of the positions function of a slider

Fluctuations a slider in a dwelling zone



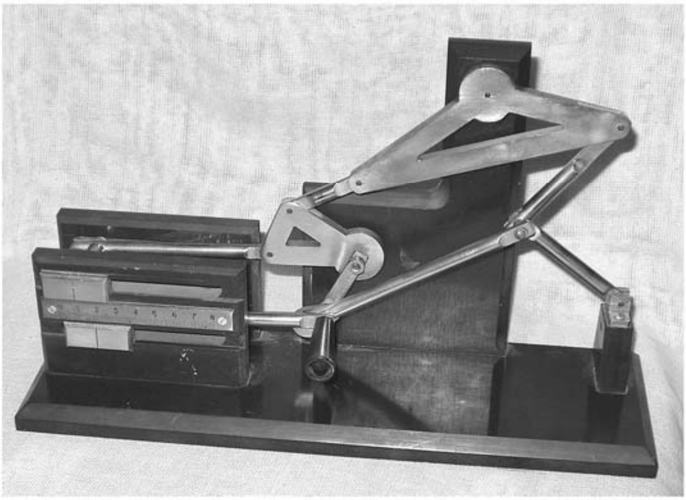


Fig. 3. 12: The model of a drawing press mechanism

In the mechanism, the scheme of which is represented in Fig. 3.10, it is possible to use axes of all parts for spindles, except for an axis of an intermediate wheel as it rotates in the opposite direction. In a mechanism of parallel cranks, it is easy to change distances between centres of apertures whereas in mechanism with cogwheels it is difficult to do.
A photo and type diagram of the mechanism of a finger conveyor is given in Fig. 3.11. Movement from camshaft 1 is transferred to rockers 2 and 7. The first one provides platform 5 movement in the horizontal direction, the second one – in the vertical direction capturing platform 5 moves upwards, then it moves horizontally. At the end of this moving it stops and starts to move downwards (releasing details), and then moves horizontally, coming back to the starting position. Usage of spherical pairs and the highest pair Y essentially reduces the number of redundant constraints in the mechanism. Thus there are local mobilities: rotation around the axes of levers 4 and 9 and roller 11 which does not render a negative influence on the work of the mechanism.

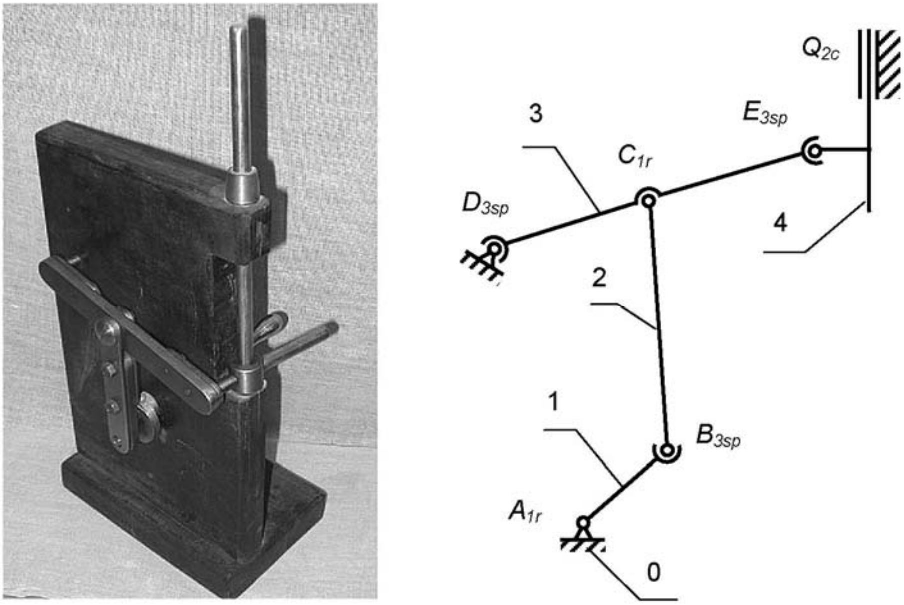


Fig. 3.13: The model of a five-bar linkage

The model of the drawing press mechanism was manufactured by a group of the Kuncevsky faculty of BMSTU students under the direction of A. Golovin (Fig. 3.12). The model represents a planar linkage with eleven mobile links. The model has one input link - crank 1 and two output – sliders 3 and 11. Movement from the input link 1 is transferred to couplers 2 and 4. Coupler 2 is connected to the output by slider 3. This slider carries out the operation of drawforming. Coupler 4 is connected with slider 11 by a kinematics chain the consisting of links 5–10. Slider 11 carries out the operation of clamping the blank and its confinement during drawforming.
The model of five-bar linkage, represented in Fig. 3.13, does not contain redundant constraints and local mobility. The mechanism is intended for transformation of rotary

movement of a crank (1) in the reciprocating motion of slider 4. If such a mechanism only has one-mobile (revolute and prismatic) pair, the number of redundant constraints will be equal to seven

$$q = W_0 + W_1 - W = 1 + 0 - (6\cdot4 - 5\cdot7) = 7,$$

where W_0 is the set mobility of the mechanism, W_1 is the local mobility in the mechanism and W is the calculated mobility of the mechanism. For elimination of these redundant constraints, three spherical and one cylindrical pairs are used, then

$$q = W_{0\,+}W\ - W = 1 + 0 - (6\cdot4 - 5\cdot2 - 4\cdot1 - 3\cdot3) = 0.$$

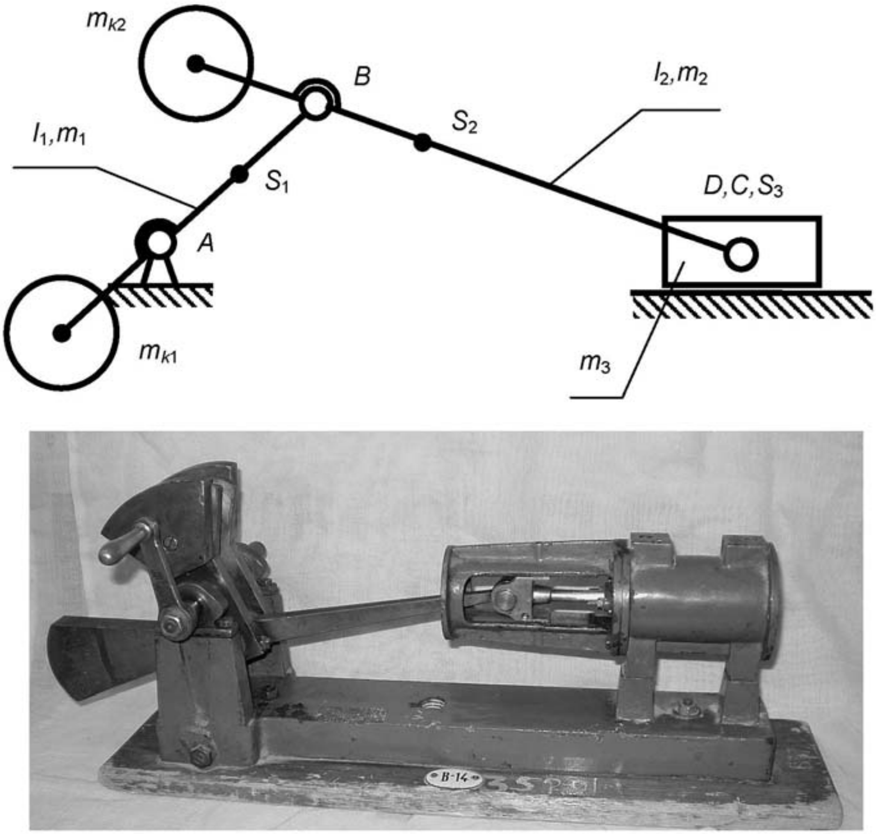


Fig. 3.14: Model of a statically counterbalanced crank-slider mechanism of a piston compressor

Model B-14 (Fig. 3.14) represents a statically counterbalanced slider-crank mechanism of a piston compressor. The model was manufactured in the TMM department. The basis of the model is a one-cylinder piston compressor for the demonstration installation developed and manufactured by SouzVuzPribor (Special Design Office) in the 1960s of the last century. This installation was intended to demonstrate of fluctuations of the compressor's basis and elimination of these fluctuations after equilibration (installation of counterbalances on the links of the mechanism). The model evidently shows an increase in masses and gabarits of the mechanism at its full equilibration.
It is used for demonstration during lectures on the equilibration of mechanisms.

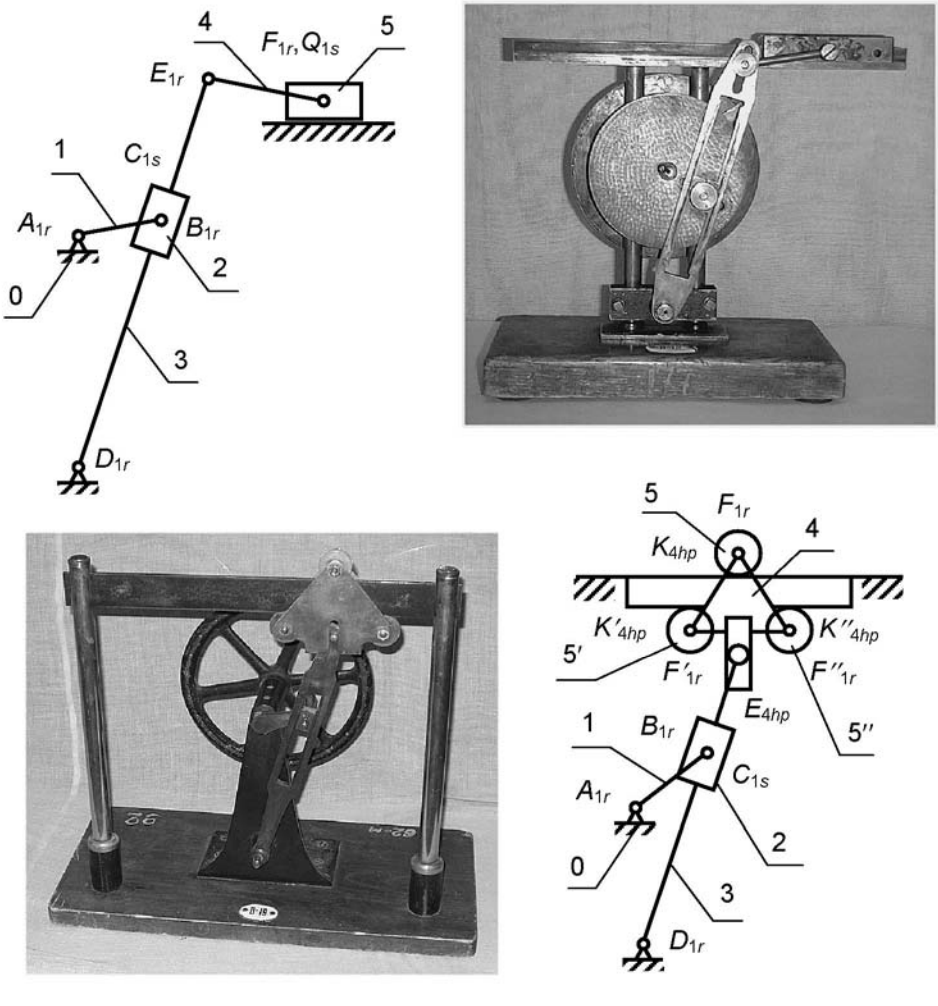


Fig. 3.15: Model of a six-bar mechanism with a rocking link

Model B-18 (Fig. 3.15) is a six-bar mechanism with a rocking link. This mechanism is widely applied in planers and slotting machines, presses, pumps and other devices. It consists of five mobile links, five revolute and two prismatic kinematic pairs. When applied to one-mobile pairs the mechanism has six redundant constraints. The opportunity of changing the pair's *E* fastening on link 3 that allows the reduction of the angle of pressure between coupler 4 and slider 5 is stipulated. The model was designed in the TMM department and manufactured in its educational workrooms. It is used for demonstration during lectures on linkages.
Model B-19 represented in the bottom part of Fig. 3.15, is also a coulisse linkage which transforms the rotary movement of a crank 1 to the translational movement of slider 4. In this mechanism, link 3 is connected with link 4 by highest four-mobile pair *E,* and link 4 moves on a beam of a frame on three rollers 5, 5′ and 5″. The rollers are connected with link 4 by revolute pairs. Such a design replaces sliding friction by rolling friction in prismatic pair. There are seven mobile links in mechanism, six revolute, one prismatic and four highest pairs. Mechanism has one basic mobility, three local mobilities (rotation of rollers 5 around their axes) and four redundant constraints (three in contour *ACD* and one in a chain of geometrical closing of the highest pairs).

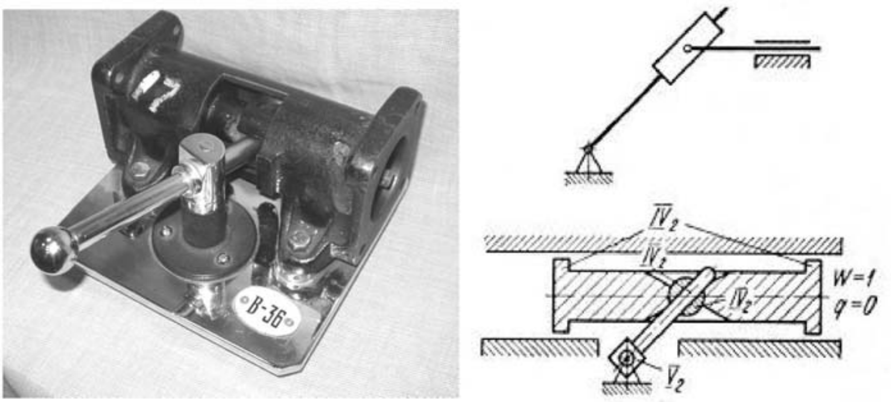


Fig. 3.16: Model of a cross-slide mechanism

One version of four-bar linkages is the cross-slide mechanism. The model of such a mechanism and its scheme are represented in Fig. 3.16. Such a design of the mechanism gear was offered by A. Ivanov [31]. The mechanism is applied in reverser drives, brake switches and switches of motor coach electrical potential. A real design of one such mechanisms is used for this model. In the mechanism, the following pairs are used: one revolute and three cylindrical. It has one mobility and does not contain redundant constraints.
Models of four-bar linkages represented on Fig. 3.17 (crank-slider and four bar linkage) are intended for demonstration during laboratory work on metric synthesis. Over 60 years of the last century, four complete sets of such mechanisms were manufactured in the workrooms of the TMM department. Models of mechanisms allow to change

lengths of links and to measure the angular and linear positions of input and output links. In laboratory work, calculations of link lengths are made using set conditions. Sizes are established for each model of the mechanism.

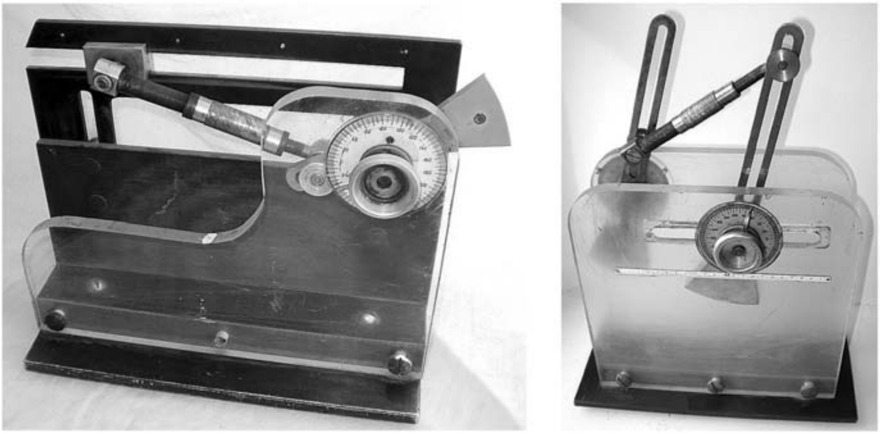


Fig. 3.17: Models of four-bar linkages

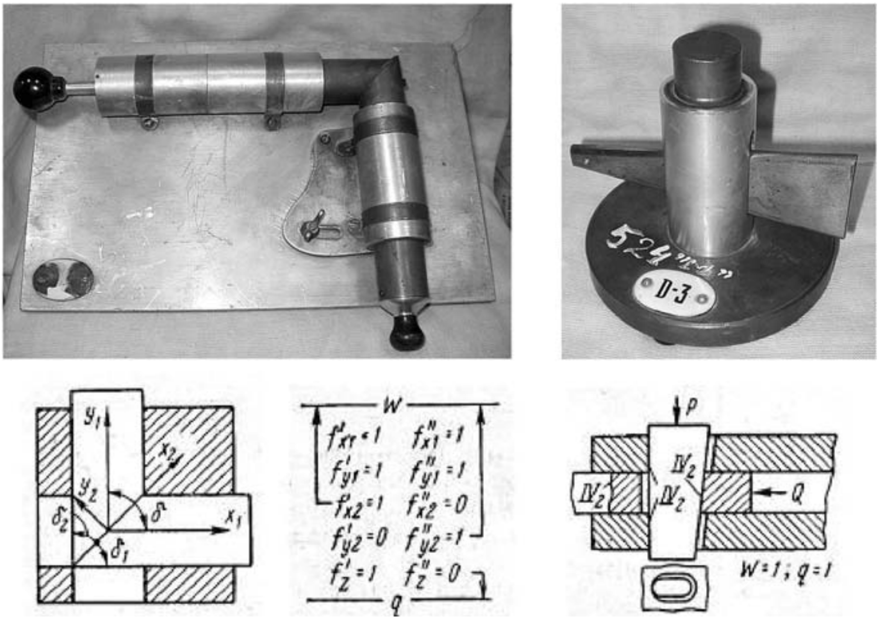


Fig. 3.18: Models of wedge mechanisms without redundant constraints

Then angular coordinates are set on input and output link coordinates are fixed. The function of the position of the mechanism was constructed using received experimental data, which was compared to the set law of movement.

Two models of wedge mechanisms are represented in Fig. 3.18. A wedge mechanism consists of two mobile links and three kinematics pairs. In the scheme of the mechanism represented in the figure on the left, pairs formed by links with a frame are two-mobile cylindrical. Links are connected among themselves by a three-mobile flat pair. The mechanism has one mobility and does not contain redundant constraints

$$q = W_0 + W_1 - W = 1 + 0 - (6\cdot 2 - 4\cdot 2 - 3\cdot 1) = 0.$$

In the second mechanism, all three pairs are cylindrical. It has one mobility and one redundant constraint which demands the exact coordination of the angle on wedge with the angle on a rod

$$q = W_0 + W_1 - W = 1 + 0 - (6\cdot 2 - 4\cdot 3) = 1.$$

3.2.2. “Rational” linkages

The next, considerable section of the model collection is rational or selfplaced mechanisms. There are planar and spatial linkages, multi-row and planetary gears, mechanisms of universal joints (Cardano mechanisms) among these models. The classes of kinematic pairs and their arrangement are chosen so as to avoid the occurrence of redundant (passive) constraints or local mobilities in the closed contours of mechanisms.

Redundant constraints appear only if links of the mechanism form closed contours (Fig. 3.19). Redundant constraints are not formed in open or closed kinematic chains. When moving an object freely, redundant constraints are absent in the manipulator. But if the manipulator performs an assembly operation, for example, inserts a dowel into an aperture, as soon as the dowel forms a pair with a link in which the aperture is executed, the kinematic chain becomes closed and the redundant constraints appear in it. Redundant constraints change the kinematic chain of the mechanism to being statically indeterminate. When there are redundant constraints, movement in a contour of the mechanism is possible: at exact performance of all sizes, due to clearance in kinematics pairs and due to the compliance of links. For a redundant manipulator constraints mean that movements of all drives are interconnected. An error even in one drive, can lead to seizure or breakage of the mechanism. Manipulators are a special variety of mechanisms, but the majority of mechanisms are formed by a closed kinematic chain. The detection and elimination of redundant constraints are actual problems for them.

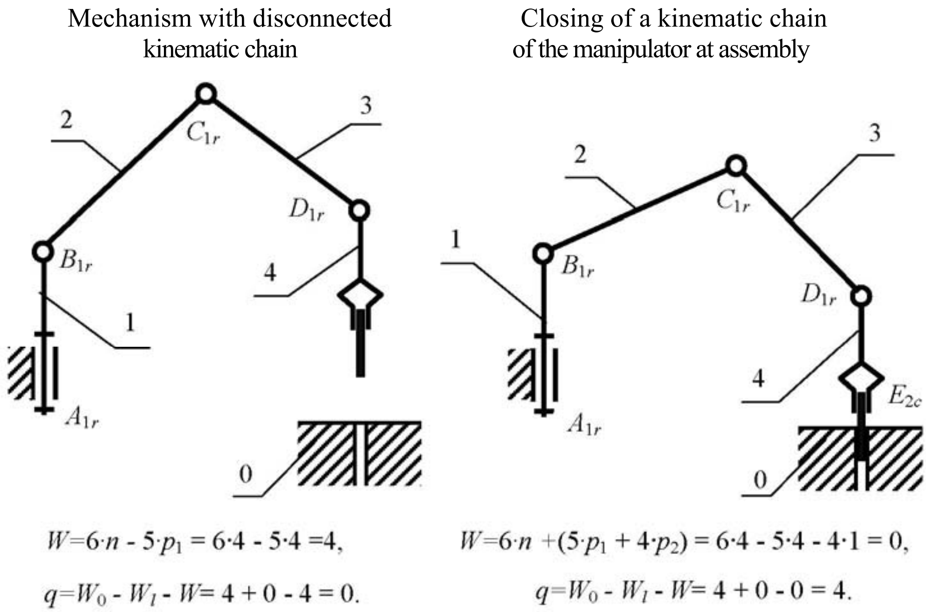


Fig. 3.19: Change of structure of the manipulator assembly operations (when closing a kinematic chain)

The number of independent contours in a mechanism is evaluated using the formula of H. Gohman [35]

$$k = n - p,$$

where

k – the number of independent contours in the mechanism
n – the number of moving links of the mechanism
p – the number of kinematic pairs in the mechanism.

To eliminate redundant constraints, beside the basic mobility, there should be mobilities in all six coordinates in each contour: three linear and three angular ones. It is possible to determine the number of redundant constraints in a mechanism (or in a contour of a mechanism) from the Reshetov – Ozol formula

$$q = W_0 + W_l - W_c ,$$

where
q – the number of redundant constraints
W_0 – the given mobility, W_l – the local mobilities
W_c – is the designed mobility of the mechanism
Local mobility of a mechanism is such a mobility that does not change the fundamental law of movement of the mechanism. Local mobilities are entered by the designer, for example, to replace sliding friction with rolling friction or to provide uniform wearing of a pair's surfaces. The mobilities of balls in a ball-bearing can be an example of local mobilities. Presence of several one-coordinate mobilities in a contour of the mechanism is a necessary (but insufficient) condition for the appearance of local mobilities. Such mobilities will be local only if they don't influence the function of the mechanism's position. Detection of redundant constraints and local mobilities can be carried out by the analysis of mobilities method [31] which is carried out for all contours of the mechanism.
A table of mobilities on six coordinates for each independent contour of the mechanism is made. The important rule of the method is that it is possible to replace linear mobilities by angular ones if their axes are perpendicular to the axis of this angular mobility. For example, z-axis angular mobility can replace x-axis or y-axis linear mobility. Inverse replacement of linear mobility by angular mobility is impossible. Application of the method will be illustrated by a six-bar linkage example. The scheme of the mechanism is represented in Fig. 3.20.
The mechanism consists of five mobile links and seven kinematics pairs. There are two independent contours in the mechanism. The first contour includes pairs MQED, the second one includes pairs ABCD. It is necessary to note that both of the contours include pair D. Change of its mobility will influence both of the contours. The first contour will be discussed.
There are four mobilities in it: three x-axis rotary ones and one y-axis linear mobility. One of three rotary mobilities provides the basic mobility in a contour, the second one replaces x-axis linear mobility. Thus, there are three redundant constraints in the first contour. It is necessary to change classes of kinematics pairs to eliminate these redundant constraints. For example, to replace one-mobile revolute pair Q by a three-mobile spherical one adding angular x-axis and y-axis mobilities, and to replace the revolute pair *E* by a two-mobile cylindrical one, adding z-axis linear mobility. There will be no redundant constraints and local mobilities in the first contour after such changes. The second contour has two redundant constrains: z-axis linear and x-axis angular ones. The z-axis linear mobility in pair *B* and x-axis angular mobility in pair A are added for their elimination.

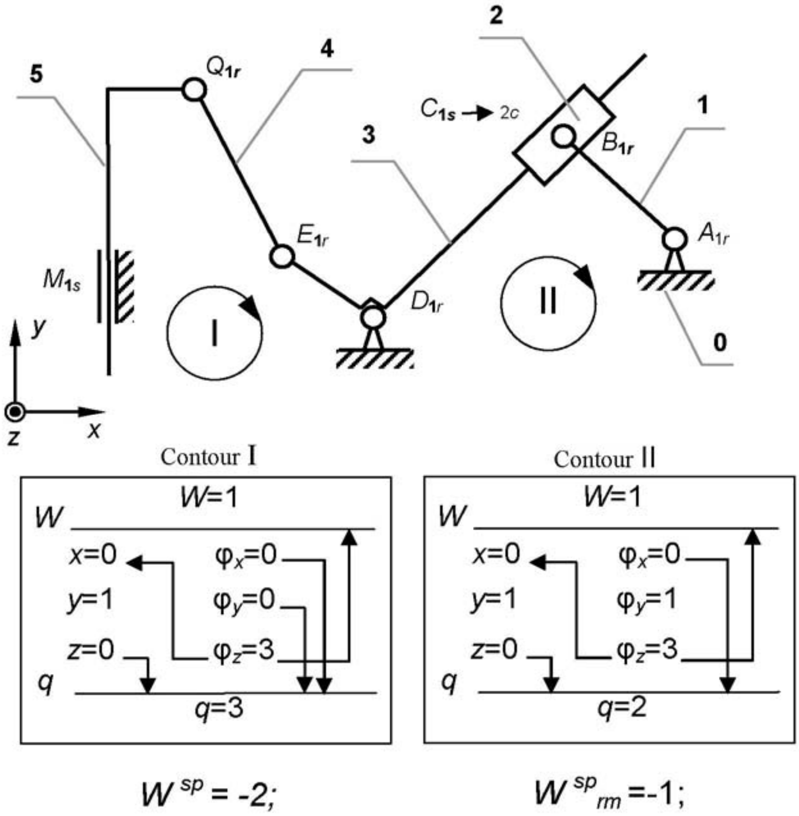


Fig. 3.20: Structural analysis of the linkage by the analysis of mobilities method

Finally (Fig. 3.21), the number of mobilities in the derived rational mechanism is

$$W^{sp} = 6 \cdot 5 - (5 \cdot 2 + 4 \cdot 4 + 3 \cdot 1) = 30 - 29 = 1,$$

and the number of redundant constraints

$$q^{sp} = W_0 + W_l - W^{sp} = 1 + 0 - 1 = 0.$$

However, we shall return to the models of rational linkages from the collection of the BMSTU TMM department. First, the crank-slider mechanisms, four-joint linkages and coulisse mechanism will be discussed. There are 22 schemes of crank-slider

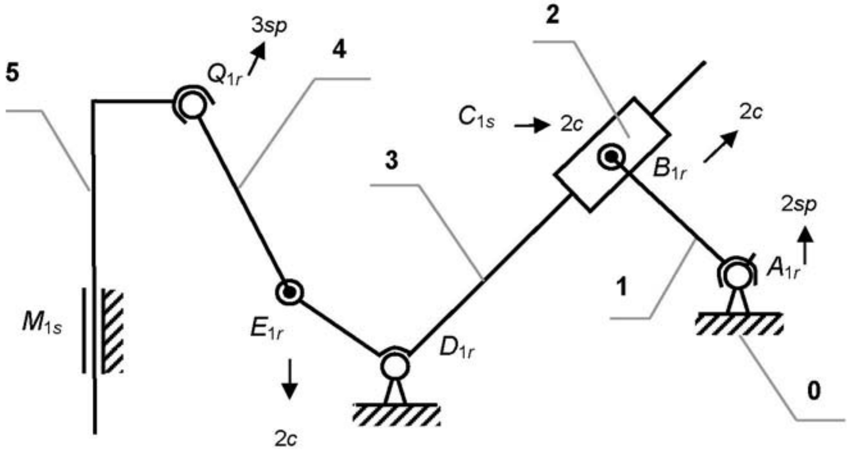


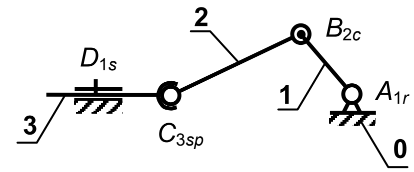
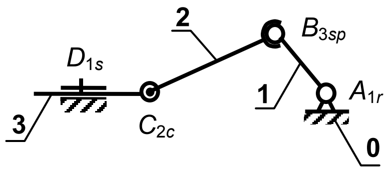
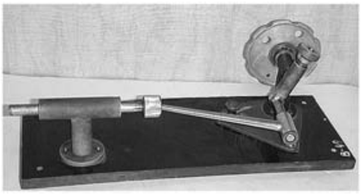
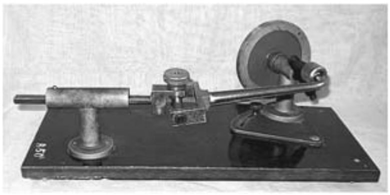
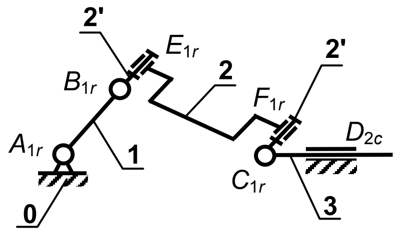
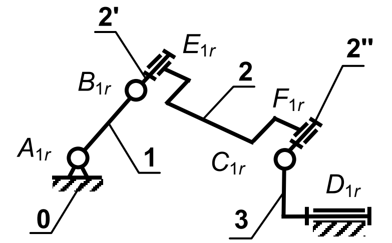
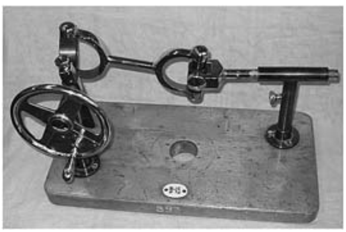
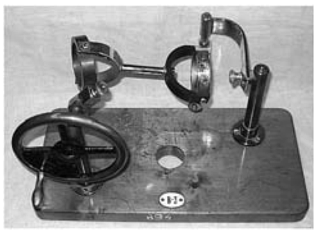
Fig. 3.21: The scheme of the six-bar linkage after elimination of redundant constraints

mechanisms, nine schemes of four-joint mechanisms and five schemes of coulisse mechanism offered in Reshetov's work [31]. A part of these mechanisms is realized in the models. Photos of these models are shown in Table 3.5. In these mechanisms redundant constraints are eliminated by a decrease in the class of kinematic pairs (reduction of the number of constraints in a pair), or by the introduction of equivalent kinematic connections made up of additional links and revolute kinematic pairs. Two crank-slider mechanisms are shown in the first row of Table 3.5. A crank-slider mechanism has one contour with three redundant constraints: x-axis and y-axis angular constraints and a z-axis linear constraint. In both mechanisms, one revolute pair is replaced by a spherical pair (x-axis and y-axis angular mobilities are added) and another one by a cylindrical pair (z-axis linear mobility is added) for the elimination of redundant constraints. In the mechanism on the right, the spherical pair is located in joint *B,* and the cylindrical one in joint *C.* On the contrary, in the mechanism on the left the cylindrical pair is located in *C,* and the spherical one in *B*.

The mechanisms in which redundant constraints are eliminated by the introduction of kinematic connections are located at the bottom of the table. On the left, there is a crank-slider mechanism. This mechanism has one contour in which there are three redundant constraints: x-axis and y-axis angular constraints and a z-axis linear constraint. For elimination of constraints, two additional links ($2'$ and $2''$) and two *y*-axis oriented revolute pairs *E* and *F* are entered. Each link adds six mobilities to the mechanism, and each revolute pair imposes five constraints on the relative movement of links. Thus, two y-axis angular mobilities are entered in addition to the contour of the mechanism. Besides, prismatic pair D is replaced by a two-mobile cylindrical pair.

The spatial four-bar linkage is pictured on the right. The kinematic connections are used here too; two links and two y-axis oriented revolute pairs *E* and *F* are added. However, in this mechanism, only two of the three redundant constraints are eliminated. Distinctions of this mechanism are that only revolute kinematics pairs are used in it, and movement is transmitted between shafts where axes are crossed at right angles.

Table 3.5. Models of rational four-bar linkages

<table>
<tr><td rowspan="2">Mechanisms with kinematic pairs of various classes</td><td></td><td></td></tr>
<tr><td></td><td></td></tr>
<tr><td rowspan="2">Mechanisms with kinematic connections</td><td></td><td></td></tr>
<tr><td></td><td></td></tr>
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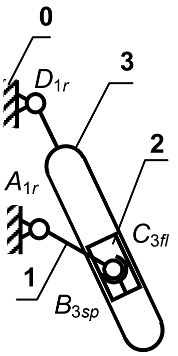
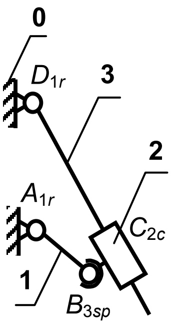
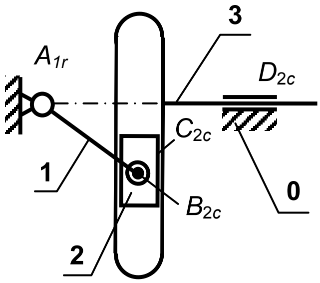
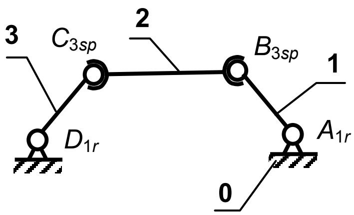
The schemes and photos of four models of four-bar linkages with rational structure are presented in Table 3.6. There are two linkages in the top row of the table. The contour of the coulisse mechanism has three redundant constraints: z-axis linear, x-axis and y-axis angular pairs. In the mechanism on the left, the revolute pair B is replaced by three-mobile spherical, and prismatic pair C by a three-mobile flat pair.

Table 3.6. Models of rational four-bar linkages

Mechanisms with kinematics pairs of various classes				
Mechanisms with kinematics pairs of various classes				
Mechanisms with kinematics pairs of various classes				

Two angular mobilities on axes x and z are added in the first pair, one x-axis angular mobility and one z-axis linear mobility are added in the second pair. Redundant constraints are eliminated at all such choice of pairs in the mechanism, but there appears to be a local x-axis angular mobility.

The second coulisse mechanism has another combination of kinematics pairs. Pairs connecting input and output links to the frame are revolute as in the previous mechanism. The pair connecting a crank to a die block is a three-mobile spherical one, and the pair placed between the die block and the coulisse is a two-mobile cylindrical pair. In work [31], it is posited that this mechanism has no redundant constraints, but it does have them. During the elimination of redundant constraints, revolute pair *B* has been replaced by a spherical one (angular mobilities on axes x and *y* are added), and the prismatic pair *C* has been replaced by a cylindrical pair (x-axis angular mobility is added). One z-axis linear redundant constraint remained and a local x-axis angular mobility appeared in the mechanism.

The model of sine-linkage is shown in the second row of the table. The z-axis linear mobility in pair *B,* the y-axis angular mobility in pair *C* and the x-axis angular mobility in pair *D* are added for the elimination of redundant constraints. There are no redundant constraints and local mobilities in the mechanism with such a choice of kinematic pairs.

The model of the mechanism of a steering trapezium is shown in the last row of Table 3.6. This mechanism is widely used in many vehicles. In this mechanism, pairs *A* and *D* are revolute, pairs *B* and *C* are three-mobile spherical. There are no redundant constraints in the mechanism, but there is one local mobility there. It is rotation of link 2 about its axis. However, this mobility doesn't exert any harmful influence on the working of the mechanism and, consequently, demands no elimination.

3.2.3. Universal joints

Universal joints or Cardans comprise a considerable part of the chapter on the collection of mechanisms. Many of these models are connected with the name of L. Reshetov. Other parts belong to chapters concerning Reuleaux—Voigt and Schröder — Redtenbacher's collections of models. In Chapter 2 there are models of Cardano mechanisms fabricated by Reshetov from Meccano detail. These "drafts" were the basis for further designing and the manufacturing of models for the collection at the BMSTU workrooms. In his researching into Cardano mechanisms, Reshetov pursued the aim of elimination of redundant constraints.

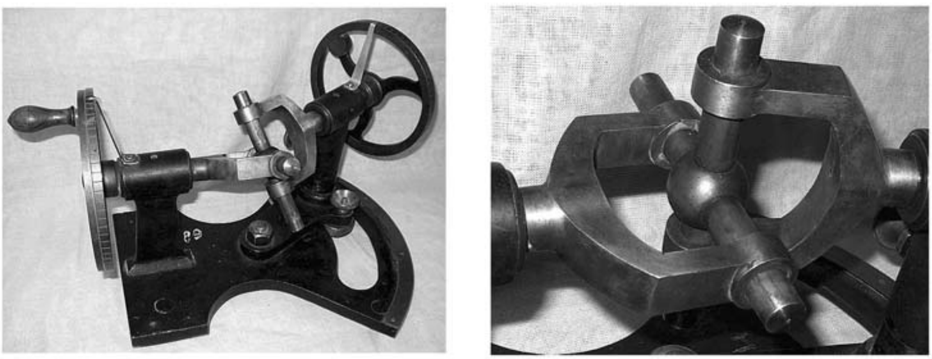


Fig. 3.22: Hooke's joint with 4-class pairs

In photo in Fig. 3.22 is pictured Hooke's joint with 4-class pairs. In this mechanism, a cross-piece is installed in forks in cylindrical pairs. When the input shaft rotates, its cross-piece is moving on an axis with a double frequency ratio. There are no redundant constraints in this mechanism but it is impossible to use it in practice. A cardan with a cube is shown in Fig. 3.23.

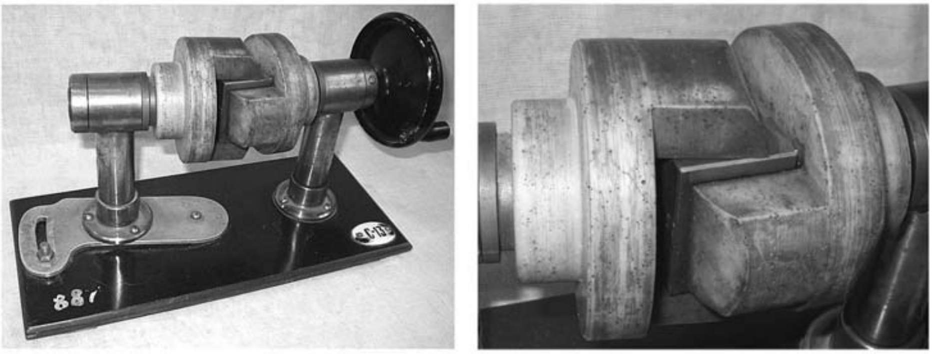


Fig. 3.23: Model of a Cardano mechanism with cube

Usually, this mechanism is used for transmission of motion between disaligned parallel shafts (Oldham clutch). However, if one makes the cube with splays and give freedom in an axial direction, then this mechanism with small chamfer angles may be used as a cardan. There are two mobilities in this mechanism: the first is the basic mobility (rotation of shafts), the second is a local one (axial movement of cube), there are no redundant constraints in the mechanism. A cardan mechanism with an intermediate cube doesn't ensure a constant transmission ratio. The coefficient of the nonuniformity of motion of this mechanism is approximately equal to the square of the angle between axes (in radian).

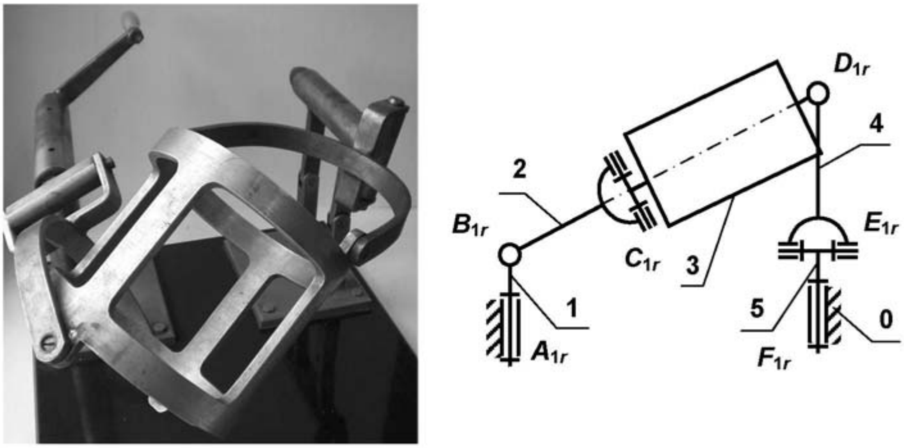


Fig. 3.24: Mixer's mechanism of the "drink barrel"

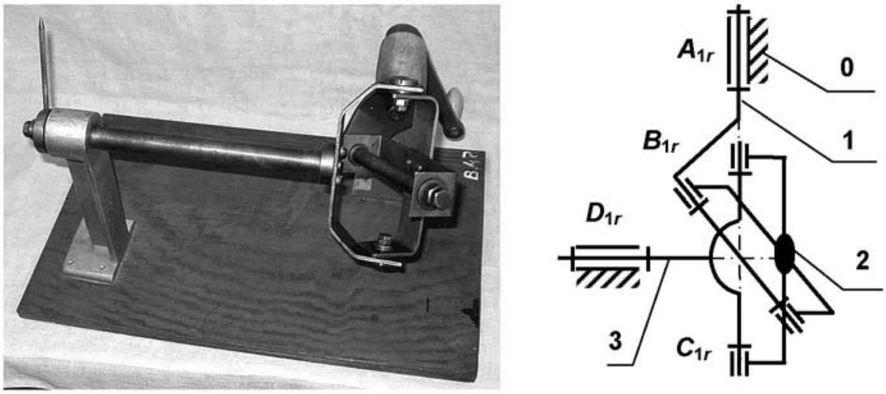


Fig. 3.25: The mechanism with an oscillating movement of the output link

In 1966, the Mixing machine “Turbula” of the Swiss firm Vili A. Bahoven (VAB) [31] was shown at the International Chemical Exhibition in Moscow, and there was an unusual cardan mechanism in its base.
In this mechanism the axes of the shafts were parallel to each other, that is the angle between the axes was 180°. The cardan cross-piece was made as a space frame with a drum with miscible materials inside it. A complicated spatial movement, the frame of which rotates the input shaft, provided good mixing. A model of this mechanism was manufactured due to Reshetov at the workrooms of the TMM department. A photo of this mechanism and its type scheme are shown in Fig. 3.24.
Another model of spatial linkage converts the rotary motion into a rotary oscillating motion. This mechanism is also based on Hooke’s joint. Axes of input and output links cross, and the angle of intersection of axes is 90º. A type scheme of the mechanism and its photo are shown in Fig. 3.25.
The collection also includes several models of little known constructions of cardan mechanisms. In Fig. 3.26 a mechanism is shown in which movement from the input shaft imparts through intermediate links.

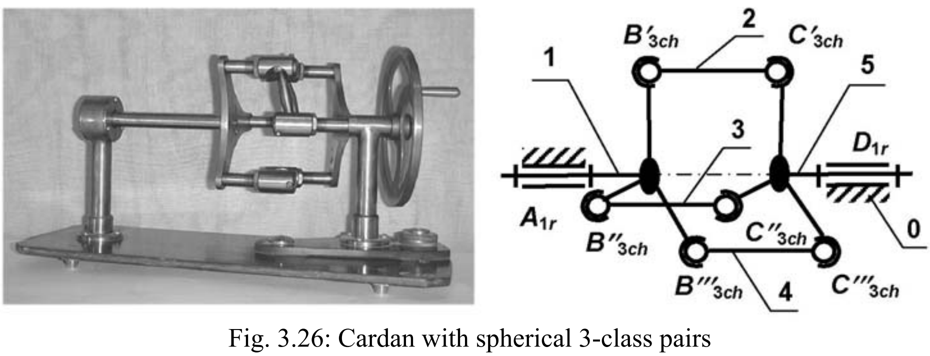


Fig. 3.26: Cardan with spherical 3-class pairs

Kinematics pairs, which connect these links with shafts, are spatial three-moving ones. There are three local movements in the mechanism: rotations of intermediate links 2, 3 and 4 about their axes.
Trackt's cardan with a split key is shown in Fig. 3.27.

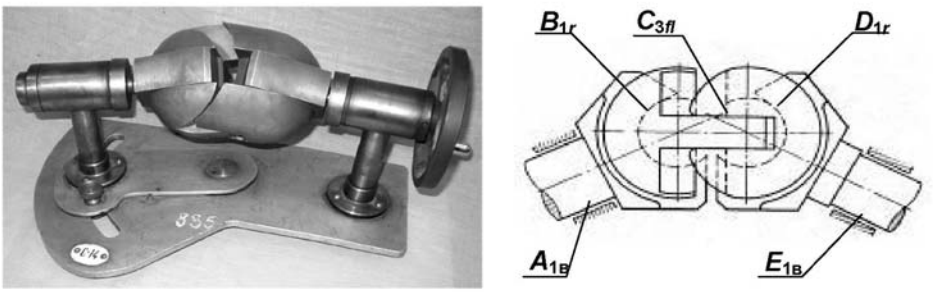


Fig. 3.27: Trackt's cardan

This cardan is analogous to Hooke's double joint in its kinematics. This mechanism has four intermediate links. Crosses of input and output links are connected to split keys by revolute pairs, split keys form a three-moving spatial pair between themselves. Trackt's mechanism is a "synchronous cardan", that is it doesn't bring additional nonuniformity to the link's movement.
Fig. 3.28 shows the cardan mechanism in [31] called a "rolling mechanism" because it is used in rolling mills for big torsion torques. The input and output links of this mechanism are connected through intermediate links – segments. These segments form a revolute pair with one of the links and a three-moving flat pair with another link. An additional link – die block – is used for fixing the driving fork. This mechanism has many redundant constraints, therefore its manufacturing requires high accuracy and special technology.

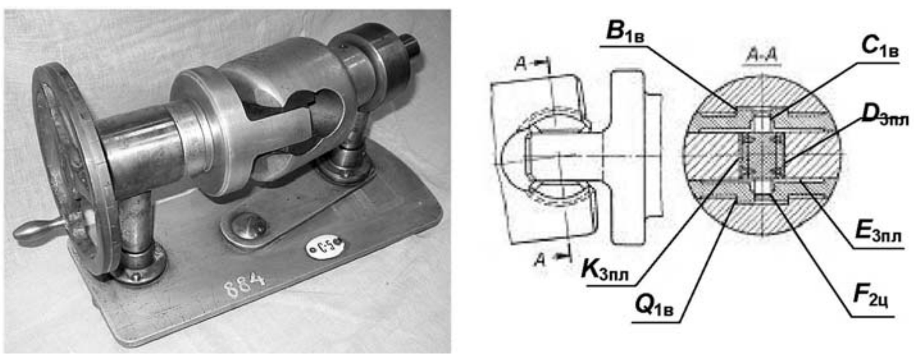


Fig. 3.28: Rolling cardan

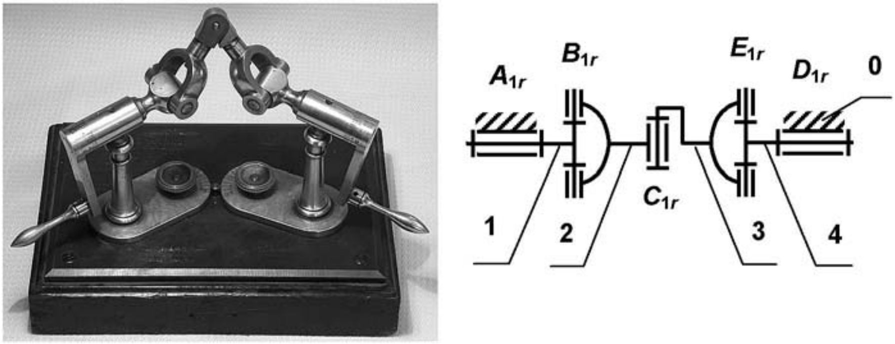


Fig. 3.29: Klemens's joint

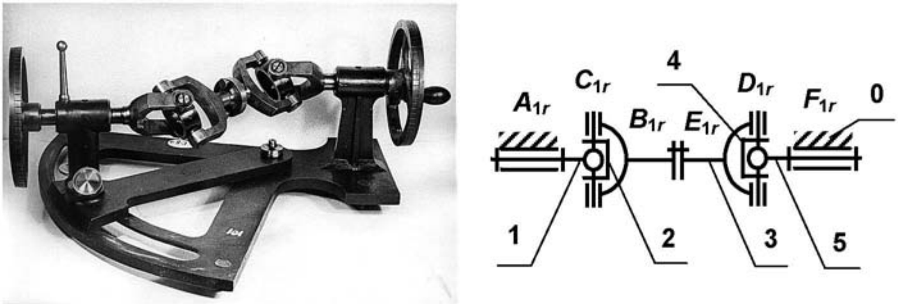


Fig. 3.30: Hooke's double

One more cardan mechanism of the TMM department is Klemens's cardan. This mechanism is described by Reuleaux in [42] and included in his collection in the form of three models of different modifications. The mechanism consists of four moving links and five revolute kinematic pairs. There is only one of these models in the BMSTU collection (Fig. 3.29).

In practice, Hooke's joints are used in pairs because only then could the uniform rotation of the output shaft be provided. To provide the uniformity, it is necessary to install forks of intermediate links in the same plane. If this condition isn't executed, the nonuniformity of the rotation of the output shaft is increased. It is demonstrated by the model pictured in Fig. 3.30.

In the collection, there are some models of ordinary Hooke's joints, which are designed for demonstrating the rotation nonuniformity of a mechanism's output shaft. Measuring angular scales which can measure the angular location of the output link by the defined angular coordinate of the input link are installed on the shafts of these models. The ability to change the angle between the axes of shafts is provided in these models. Photos and schemes of models of such mechanisms are shown in Figs. 3.31–3.33.

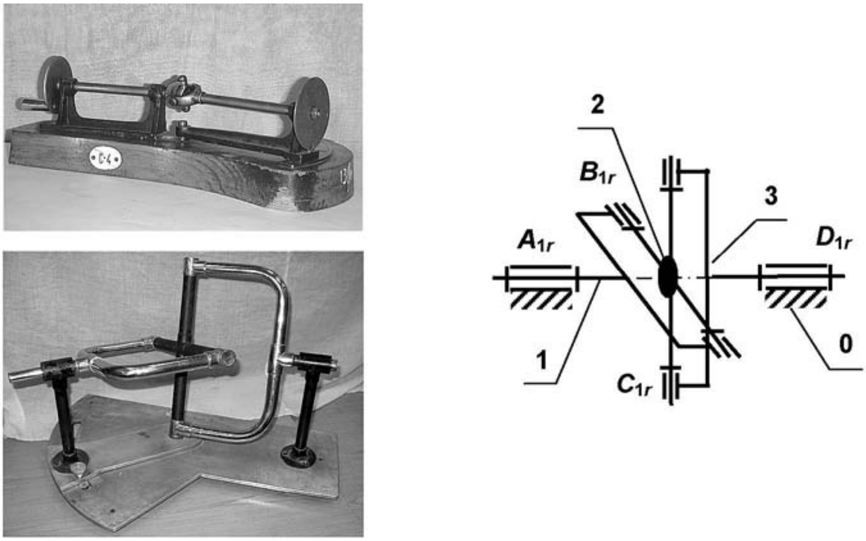


Fig. 3.31: Hooke's joint with revolute 5-class pairs

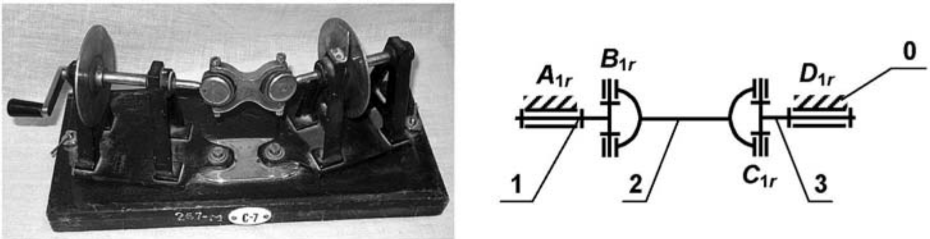


Fig. 3.32: Cardan with flat fork

Fig. 3.33: Belt cardan

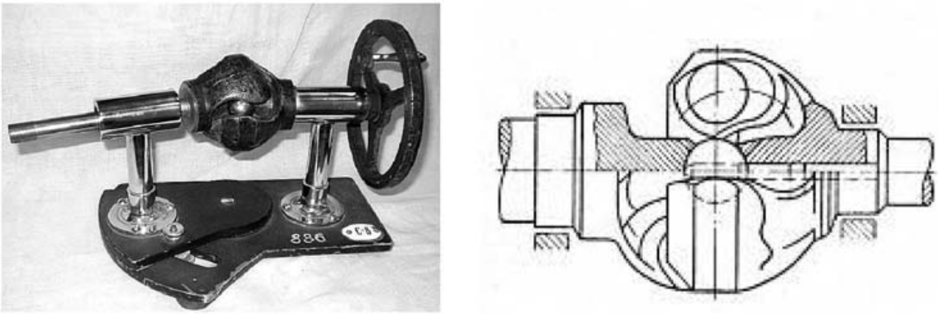


Fig. 3.34: Weiss's ball cardan

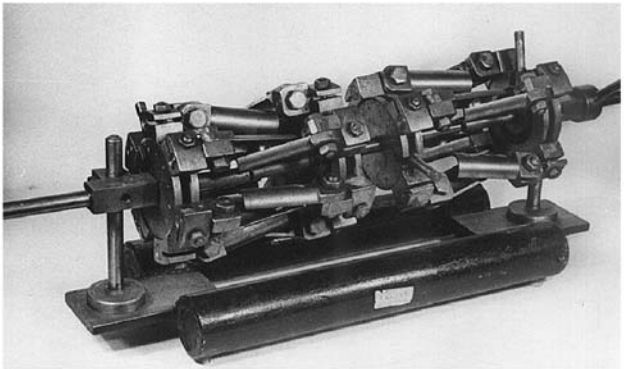


Fig. 3.35: Reshetov's ball cardan

Cardan mechanisms with higher pairs, for example ball cardans, have become more and more widespread in recent years. Nowadays, these mechanisms are presented by two models in the collection: by Reshetov's ball cardan and Weiss's cardan (Figs. 3.34, 3.35). For a long time, the use of these mechanisms was restrained by technological obstacles, which appeared when the mechanisms were being produced. With motorcar construction changing to the delivery of front-driven cars, mass-production of these cardans was mastered.

3.2.4. Hydromotor's mechanisms

Hydraulic drives are often used in various machines. In these drives the liquid carries out the function of a transfer link between the hydraulic pump's mechanism and the hydromotor's mechanism. The hydraulic drive allows the transfer of large capacities (up to 4,000 kW) and to create big loadings on output. Therefore, it is used in power drives. The collection of models includes some mechanisms of hydromotors or pumps. These devices are usually reversible. If you supply a liquid under pressure to the mechanism by means of a valve cure it will carry out hydromotor functions. The mechanism can be used as a pump when, for example, an electric motor rotates the mechanism's shaft. Various mechanisms used in hydraulic drive systems are known: gear pumps with external or internal gearing, gear pumps with internal cycloidal gearing, planar or spatial linkages with pistons or rams.

These models are spatial linkages. The first model (Fig. 3.36) represents a spatial mechanism of an axial hydromotor with its input link made in the form of “a taper washer”.

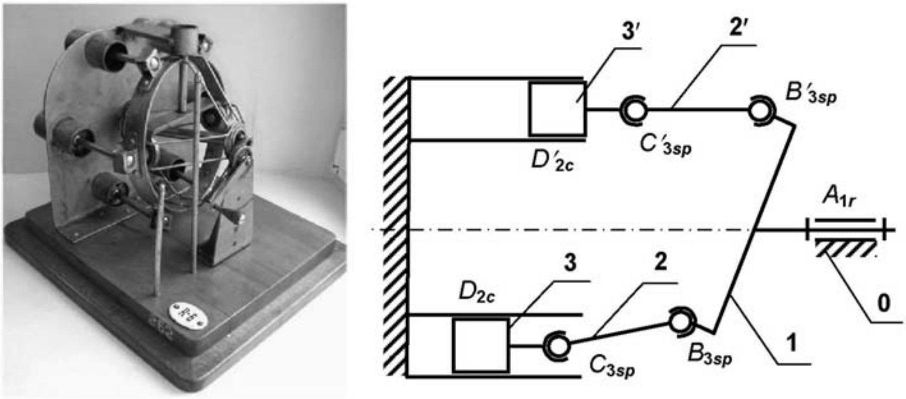


Fig. 3.36: The model of spatial axial hydromotor’s mechanism

The distinction of this mechanism is that redundant constraints are eliminated in it. There are a lot of local mobilities in the mechanism: rotation of hydraulic rams 3 about their axes, rotation of couplers 2 about their axes. However, all of these mobilities are specially entered into the mechanism to provide reduction and uniformity of wearing of the contact surfaces in pairs C and D. The model was designed by Professor L. Reshetov and manufactured at the BMSTU’s TMM department.
Two other models (Fig. 3.37) are construction units of real hydromotors made according to a given type of scheme. These models differ in design, sizes and number of hydraulic rams. The model R-7 has seven hydraulic rams. The second model has a greater size and there are nine hydraulic rams in it. The model R-7 has a wooden stand, classification and inventory numbers. The second model has neither stand nor numbers. It is simply a joint of a real hydromotor.

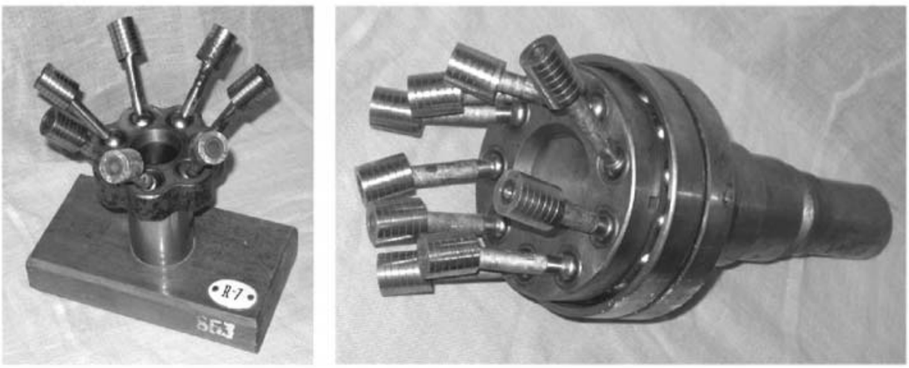


Fig. 3.37: Joints of real axial hydromotors

There are some more models of hydraulic drive mechanisms in the collection. These models are described in other chapters of this book.

3.3. Cams

Models of cam mechanisms are widely presented in the collection. They can be divided into four groups:

1. Models bought in Germany at the end of the 19th – beginning of the 20th centuries
2. Models created by postgraduate students for their dissertational works
3. Models created by employees of the department for demonstration at lectures
4. Models created by “SouzVuzPribor” for educational purposes

The three last groups will be further considered. Most of the models of the cam mechanisms are intended for demonstration in various chapters of the educational process and only a small number of models are connected with the research works of the department.

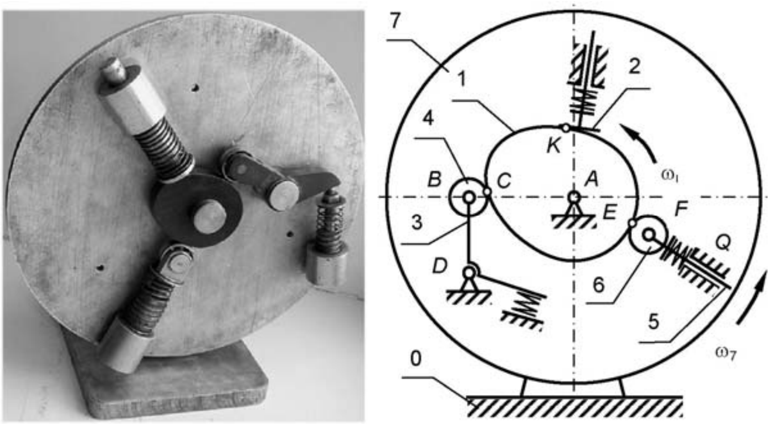


Fig. 3.38: The model of a cam mechanism with various kinds of followers

The first model (Fig. 3.38) represents the planar cam mechanism with three various kinds of followers. It simultaneously shows some kinds of cam mechanisms. Here the most widespread versions of the cam mechanisms are shown: the mechanism with a rocker follower and two mechanisms with translating followers. The rocker follower 3 is executed in the form of a bent lever and on one its end roller 4 is located and on the other end spring. Follower 2 has a flat working surface. Follower 5 has the roller 6 and its working surface is cylindrical. The force closure of higher pairs K, C and E is provided with cylindrical spiral springs. All followers are established on a disk 7. The disk can rotate around the axis and can be fixed as motionless on a frame by a screw. The shaft of cam 1 can be fixed by the screw to a frame too. Therefore it is possible to show an inverted moving on this model: the cam is fixed and the followers rotate around an axis of the cam together with disk 7. The model was designed and made in the BMSTU TMM department in the 1950s of the last century. It is a rather successful visual manual. The model of the cam mechanism with a rocker follower (Fig. 3.39) consists of the details and units of a tape drive mechanism of a accumulating device of a computer. The mechanism connects the position of the shaft of a cam with the contacts of the terminal switch 4. One phase of the distant dwell contacts of the switch

are opened but in other phases they are closed. The model was made by Ass. Professor V. Tarabarin of the TMM department in 2003. It shows an application of the cam mechanisms in the peripheral devices of the computer. The follower of the mechanism is supplied by a roller and force closure of the higher pair is the power by means of a cylindrical spiral spring. Readjustment of the moment of the contact's disconnection of the switch is made by an adjusting screw.

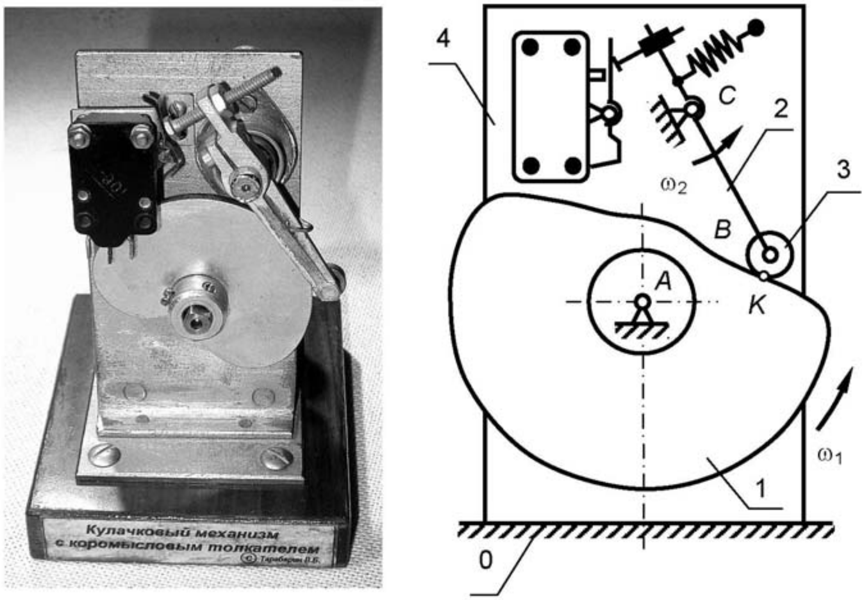


Fig. 3.39: The model of a cam mechanism with a rocker follower

In the 1960s years of the last century, the organization "SouzVuzPribor" which was engaged in the development and manufacture of the educational equipment created a complete set of demonstration models of mechanisms for the TMM course. The models of this complete set were made by "SouzVuzPribor" and faculties of universities were equipped by it. Some of these models stored in the collection of the TMM department are devoted to cam mechanisms. Model *E*-31 (Fig. 3.40) represents a cam mechanism with a slider-follower and eccentric-cam. The feature of this model is that the replacing slider-crank mechanism is located in a parallel way by a cam mechanism. Replacing mechanisms play a highly important role in the theory of machines and mechanisms. Its allow solving problems for mechanisms with higher pairs using methods developed for mechanisms with lower pairs. Even more important that it shows a generality of all mechanisms. On formation of the replacing mechanism, the higher pair is replaced by a link and two lower pairs. Thus, the length of a replacing link is equal to the distance between the higher pair's centers of curvature of profiles at the point of contact. The centers of the pairs are located at the centers of the curvature of the profiles.
In our example, the length of an additional link (the coupler) is equal to the sum of radii of a cam and a roller, the joint C is located in the center of the curvature of a cam

in point C, and the joint B is located in the center of the curvature of a roller in point B. The roller in the cam mechanism doesn't transform the movement of the output links. It's a problem to replace a sliding friction in a higher pair by rolling friction. Installation of a roller increases the number of links of the mechanism in one, and there appears an additional revolute pair B and an additional "local" mobility occurs in it. Local mobility in mechanisms is mobility which doesn't influence the basic kinematic characteristics of the mechanism and doesn't change its function of position. These mobilities are entered by the designer for, among other purposes: a change of the kind of friction, increase of uniformity of wearing of a pair's elements, etc. As in the replacing mechanism, there is no higher pair and as a roller isn't necessary, it is deleted (together with the revolute pair).

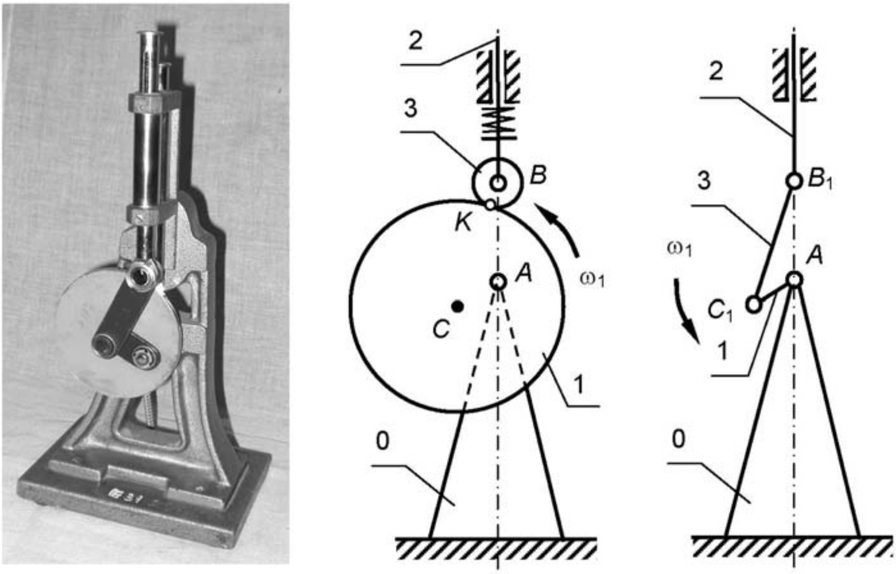


Fig. 3.40: The model of a cam mechanism with a slider-follower and eccentric-cam

The next model (Fig. 3.41) is a cam mechanism with an off-axis slider-follower. The working surface of this follower is pointed. In such a form of a follower, the constructive profile of the cam coincides with the theoretical (or center) profile. The model allows the changing of size and direction of the follower's eccentricity. The model pictured in Fig. 3.42 differs from the considered model only by the presence of the roller 3 on the follower.

Some of the following models represent spatial spherical cam mechanisms. This kind of mechanism hasn't wide proliferation because of the complexities arising during manufacture. The doubtless advantage is the opportunity of transferring movement between the axes crossed under some angle Σ. The cam of the mechanism which is shown in Fig. 3.43, represents a part of a hollow sphere. The end surface of this sphere

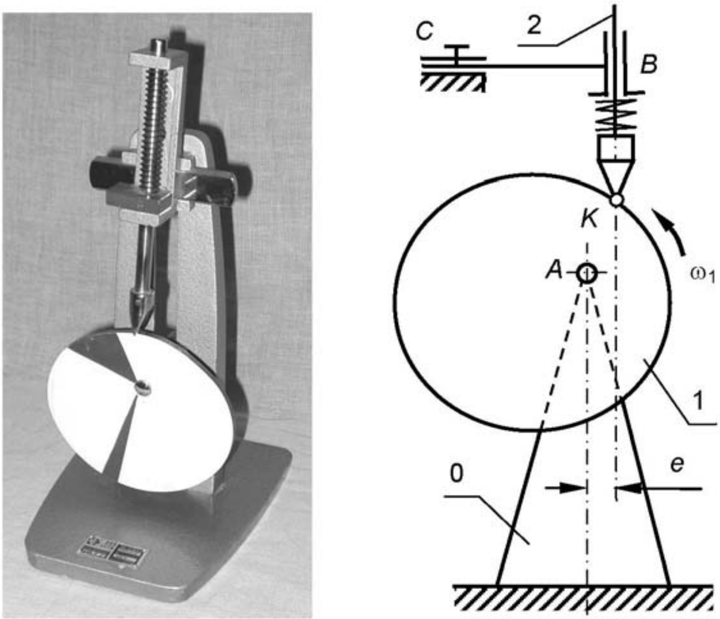


Fig. 3.41: The model of a cam mechanism with an off-axis slider-follower

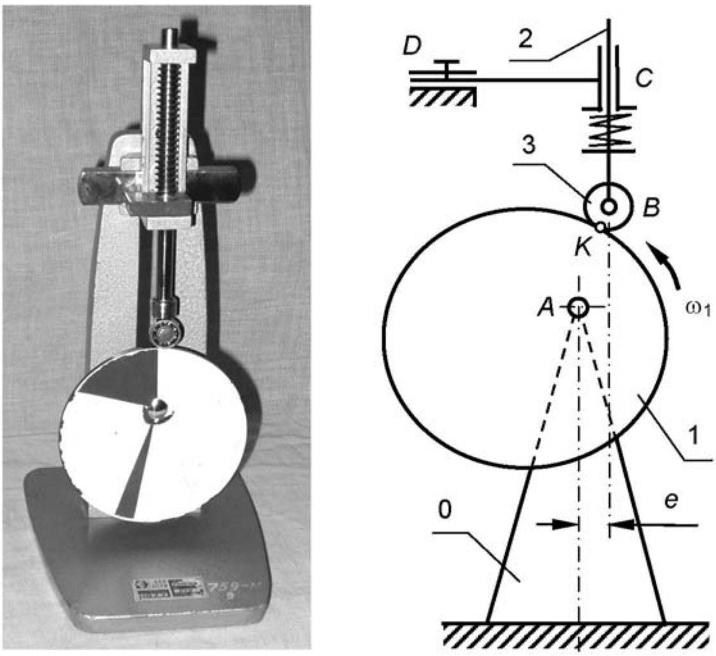


Fig. 3.42: The model of a cam mechanism with an off-axis roller-follower

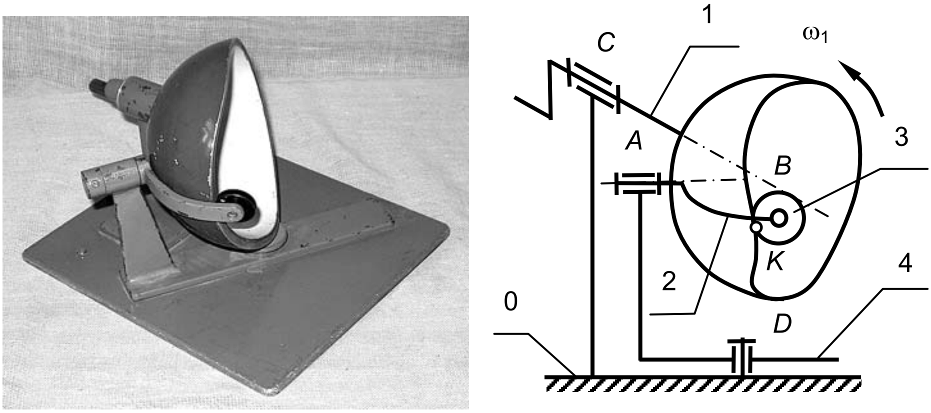


Fig. 3.43: The model of a spatial cam mechanism with a spherical cam and variable axes angle

is executed in the form of the cam's profile. The lever of the follower of cam 2 is bent in an arch the center of which coincides with the center of the sphere of the cam. The axes of the cam's shaft and the follower's shaft are crossed under Σ. It is possible to change this angle in within certain limits in the model. In position with the set angle Σ, the basis of the follower 4 is fixed by a screw. The follower of model 2 has the roller 3 on the end. Forced closure of the higher pair in the mechanism is the power and is provided by the gravity of the follower.

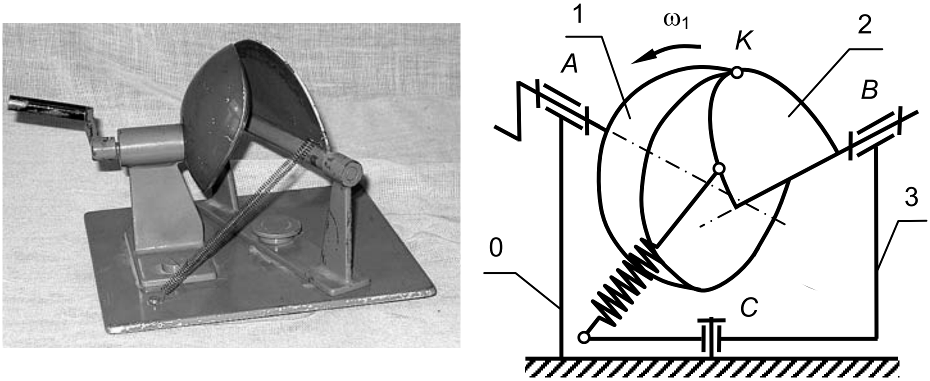


Fig. 3.44: The model of a spatial cam mechanism with a spherical cam and flat oscillating follower

The cam of the model which is shown in Fig. 3.44 represents a part of the hollow sphere too. The end surface of this sphere is executed in the form of a profile of the cam. The follower of cam 2 is executed in the form of a flat plate which is established in the frame on the bearing B. The axis of this bearing and axis of the cam's shaft are crossed under angle Σ. It is possible to change an axes angle in this model too. In

position with set angle Σ, the basis of follower 4 is fixed by a screw. Profiling of the cam in such a mechanism should be made on the condition of the concavity of the cam's profile (the plane of the follower cannot cross a working surface of the cam and, in each point, should be a tangent to it). Forced closure of the higher pair in the mechanism is the power and it is provided with a cylindrical spiral spring.

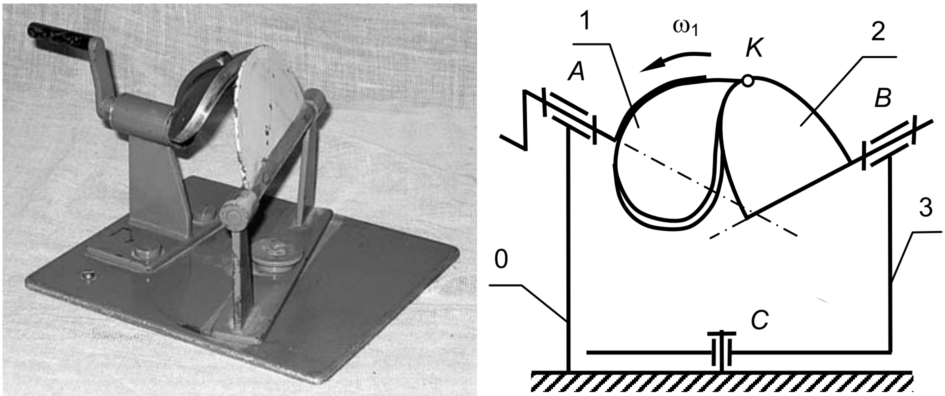


Fig. 3.45: The model of a spatial cam mechanism with a spherical cam and flat oscillating follower

The mechanism which is shown in Fig. 3.43 differs from the mechanism in Fig. 3.44 by the design of the cam. The cam is executed in the form of the small site of a hollow sphere in it. The cam 1 is displaced in relation to the cam's shaft. The shaft of the cam is connected to the cam by a lever bent on an arch of the circle. Force closure of the higher pair in the mechanism is the power and it is provided by the gravity of the follower.

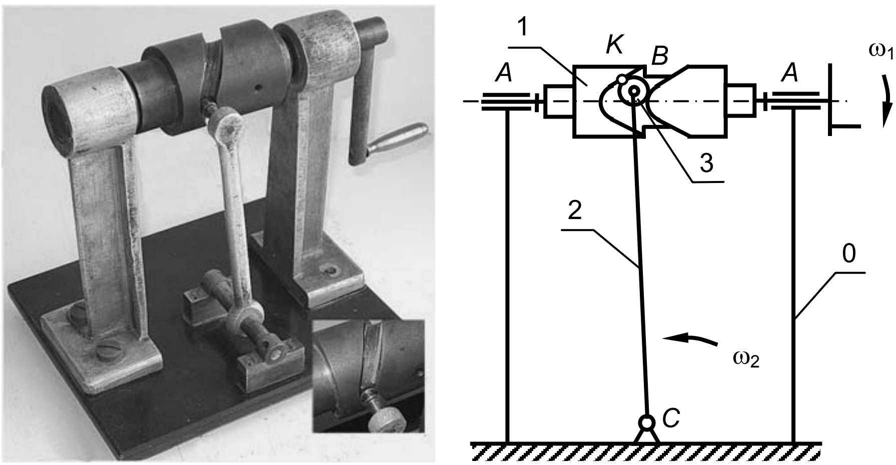


Fig. 3.46: The model of a slot cylindrical cam mechanism with a follower-rocker

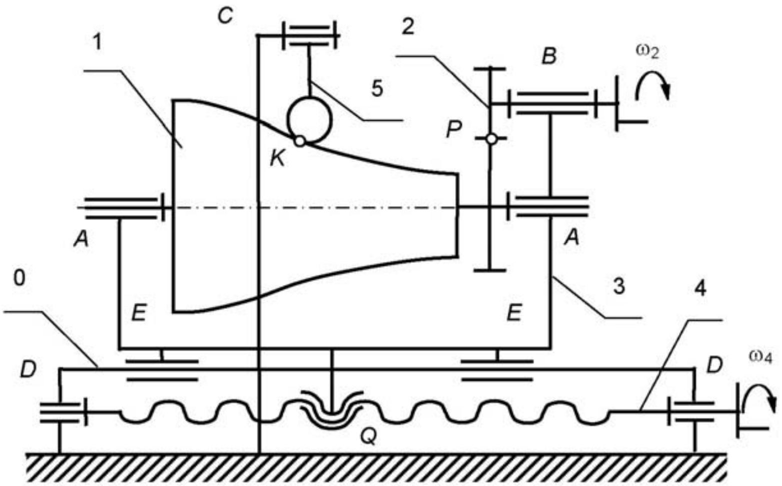


Fig. 3.47: The model of a spatial cam mechanism (conoid) with two-DOF

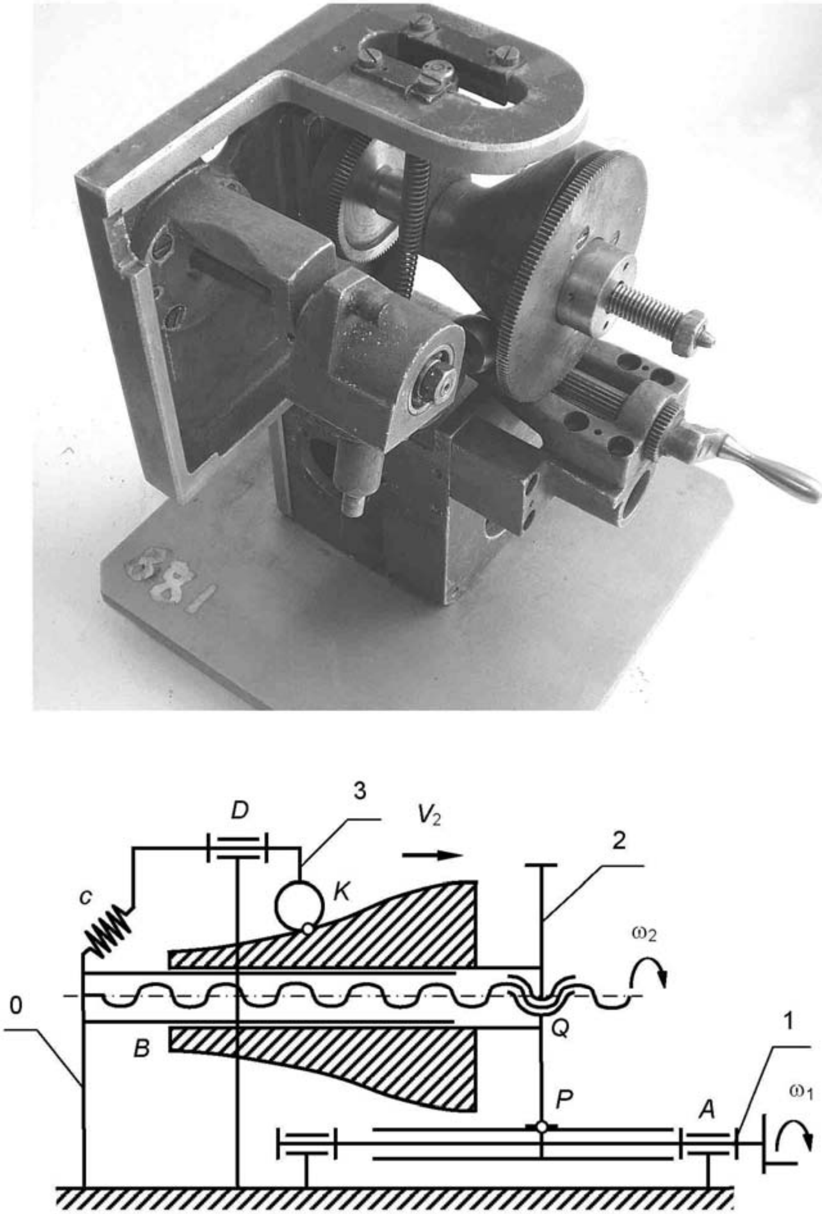


Fig. 3.48: The model of a one-DOF spatial cam mechanism (conoid)

The models (Figs. 3.40–3.45) were designed by SouzVuzPribor and made at an experimental industrial factory at the Polytechnic Institute in Karaganda in 1977. The model of the cam mechanism with a cylindrical cam attitude to spatial mechanisms was also designed by SouzVuzPribor. The axis of the input shaft (cam shaft) is crossed with the axis of the output shaft (follower shaft). The angle of crossing of the axes in this mechanism is 90°. The cam is installed on the cylinder of the input shaft and has the form of a slot of a rectangular section. The form of the slot corresponds to the demanded principle of the follower's movement. The rocker follower of the mechanism has a roller. The force, closure of the higher pair in the mechanism is geometrical and it is provided by accommodating the roller inside the cam's slot. A photo of the mechanism and its kinematics scheme are represented in Fig. 3.46. The model was made in the workrooms of the TMM department in the middle of the 1950s of the last century.

The model *E*-22 (Fig. 3.47) represents a two-DOF spatial cam mechanism. The cam of this mechanism has the form of a conoid. The profiles form some continuous set which is defined by the demanded change of the principle of the movement of the follower. To change the law of the movement of the output link it is necessary to move the cam and its related a follower in an axial direction. In the model, the cam 1 is established on a mobile platform 3. Rotation of the screw 4, through – drive "screw-nut" moves platform 3 in an axial direction. Cam 1 replays a motionless follower 5. Rotary movement is transferred to cam 1 through a cylindrical tooth gearing. A follower of the mechanism is a rocker and the working contact surface of the follower is spherical. Force closure of the higher pair mechanism is the power and is provided by the a grav ity of the follower.

The model was designed and made by collaborators of the BMSTU's TMM department at the end of the 1940s of the last century.

The model of the spatial cam mechanism (Fig. 3.48) is executed on the basis of units of the real mechanism of a control system. Unlike the previous mechanism, this mechanism has only one mobility. Rotation of the input shaft through a cylindrical gear pair is transferred to a spatial cam 2 which is installed in the case on screw pair Q. The screw of this pair is fixed in the case and the nut is fixed on a cam. At the rotation of the cam it starts to move in an axial direction. The pin 1 permit toothing with gear 2 by the full axial displacement of gear 2. The rocker follower 3 is installed in the case on bearing D and it can't move in an axial direction. A working contact surface of the follower is spherical. Force closure of the higher pair of the cam mechanism is the power and is provided by a cylindrical spiral spring.

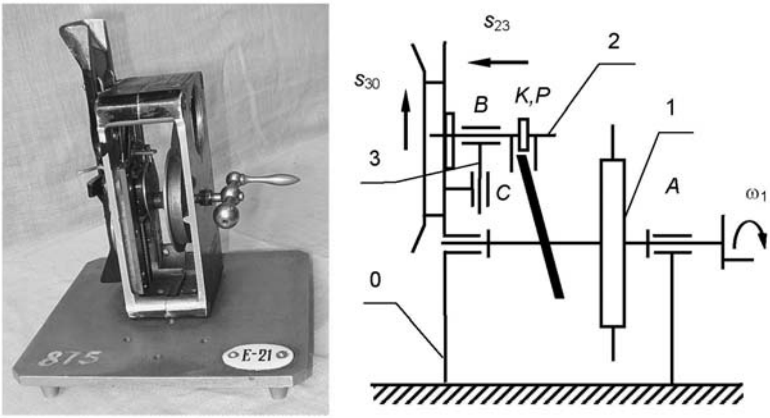


Fig. 3.49: The model of the claw mechanism of a film projector with a drive from a spatial cam

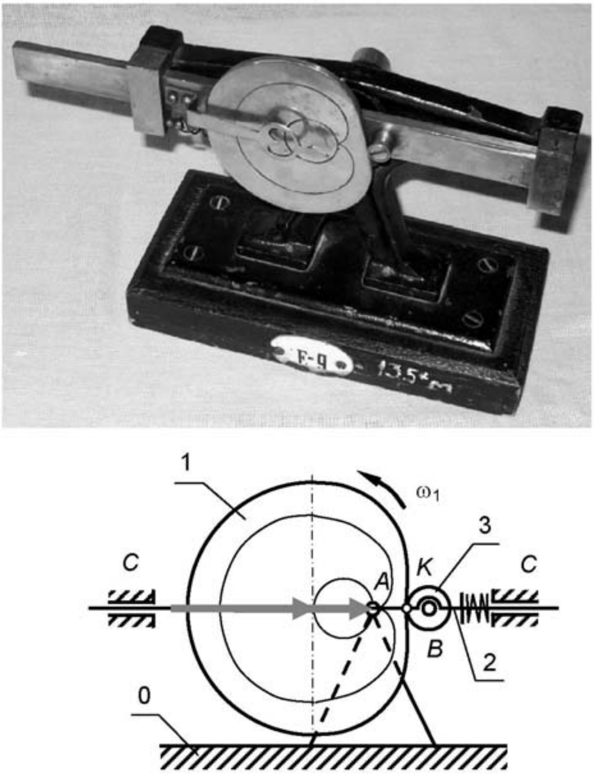


Fig. 3.50: The model of a cam mechanism with horizontal moving of follower from spatial cam

The model *E*-21 (Fig. 3.49) represents the claw mechanism of a film projector with a drive from a spatial cam. The claw mechanism carries out step-by-step film motion. The follower 2 makes two interconnected movements: moving across s_{23} and moving on a vertical 5_{30}.
Horizontal moving enters the tooth of the claw mechanism into the aperture of the film's perforations. Vertical moving traverses a film one exposure at a time. The spatial cam 1 is executed in the form of a bent plate. The form of the bend of a plate provides the demanded low the axial (horizontal) motion of the follower. The external surface of the cam is designed under the reassigned law of vertical motion of the follower. The working surface of the follower in pair *K* (at vertical moving) is cylindrical, force closure of higher pair *K* is the power and is provided by the a gravity of the follower. Horizontal movement is transferred to the follower through two pins covering the end surfaces of the cam from two sides. Force closure is geometrical in this higher pair *P.* The fly-wheel which is structurally combined with an obturator is established on the shaft of a cam. The flap of the obturator blocks the gate aperture at the moment of the vertical motion of the film.
The model *E-9* (Fig. 3.50) is a copy the cam mechanism from the Reuleaux-Voigt collection (model 8 part two catalog [41]). The model was made in the USSR in the middle of the 1930s of the last century. The model represents a planar cam mechanism with a follower with a roller moving forward horizontally. Force closure of the higher pair a roller-cam is the power and is provided by a spring. The profile of the cam is executed on the closed cycloidal curve. Relative trajectories of two points of the follower are put at the end face of the cam. One of these trajectories is a circle.
Model *E*-19 is a planar cam mechanism with a rocker follower (Fig. 3.51). It is close to the cam mechanism of the gas distribution system in the engine of a car. The follower of the mechanism has a roller. Force closure of the higher pair in the mechanism is the power and it is provided by a cylindrical spiral spring of compression. The model was designed and manufactured in the BMSTU TMM department in the 1950s of the last century.

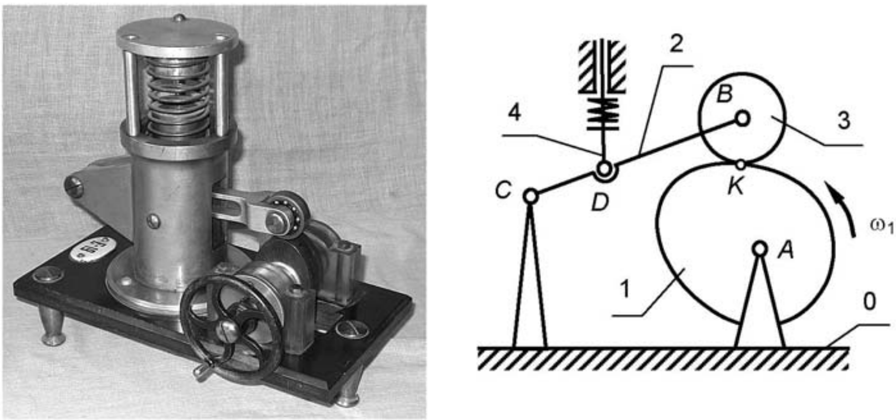


Fig. 3.51: The model of a flat cam mechanism with a rocker follower

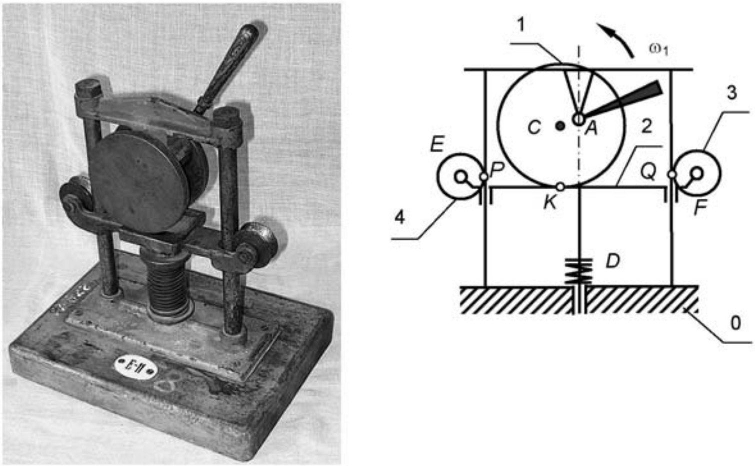


Fig.3.52: The model of a tightening device with a cam-eccentric

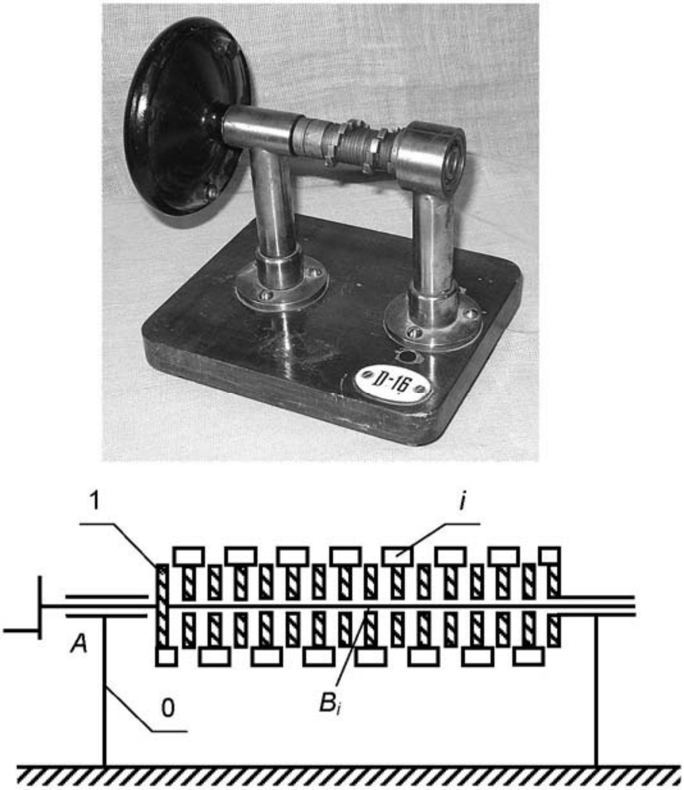


Fig. 3.53: The model of a multiway stopper

The model *E*-11 (Fig. 3.52) represents a clamping device with a cam-eccentric. There is a follower that realizes the of vertical linear motion. The follower is placed on rollers 3 and 4 in two cylindrical ways. Force closure of the higher pair in the mechanism is the power and is provided by a cylindrical spiral spring of compression. The clamp is carried out by raising the handle. The place and time of the model's manufacture is unknown.

The next model, D-16 (Fig. 3.53) is placed in the section of screw mechanisms in the classification of the TMM department but in our opinion it is necessary to place it in the cam mechanisms. The first cam is fixed on input shaft 1 and has a ledge at the right end face. The second cam is installed on a shaft 1 on the bearing *B* and can rotate around it. This cam has ledges both on the right and on left end faces. On rotation of shaft 1 the ledge of the first cam is hooked to the left face ledge of the second cam. The cams start to rotate together. Then the right ledge of the second cam is hooked to the left ledge of the third. The third cam starts to rotate too. So all *i-2* cams consistently enter into gearing. The last cam *i* has a ledge only at the left end face. This cam is fixed on the frame of the model. When the right ledge of the penultimate cam hooks to the left ledge of the last cam, shaft 1 stops. Such mechanisms are applied when it is necessary to stop shaft 1 after its turn on the set angle. The size of the angle of turn is defined by the width of the ledges and the number of cams *i.* The mechanism was designed and manufactured in the TMM department.

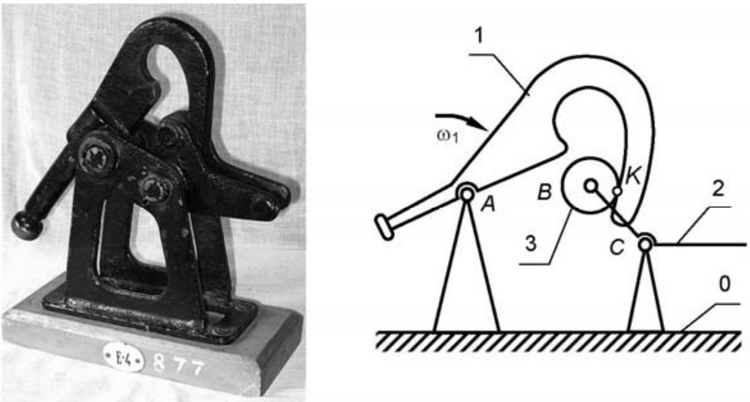


Fig. 3.54: The model of a hitch-mechanism

The model *E*-4 (Fig. 3.54) represents the real unit of the hitch-mechanism of railway carriages. This device is a cam mechanism with a rocker follower. The follower 2 is supplied by roller 3 and forms internal gearing with cam 1. The angle on which the cam can turn is approximately 90°. The profile of the cam is executed so as to provide a smooth increase in the effort of a clamp at the initial moment and then to provide reliable fixing of links with an angle of pressure close to zero. The model was manufactured in the TMM department from details of the real hitch-mechanism.

3.4. Model of toothed mechanisms

3.4.1. Simple toothing and gearings cycloid and pin gearings

The gearing model section is one of the biggest in the collection. It includes cycloidal, involutes and non-involutes, planar and spatial, simple and complicated. Let us start our review with models of simple gearings with cycloidal and pin teething. Three of them, similar to the pin gearing were manufactured in the department in 1950s and 1960s of the last century.

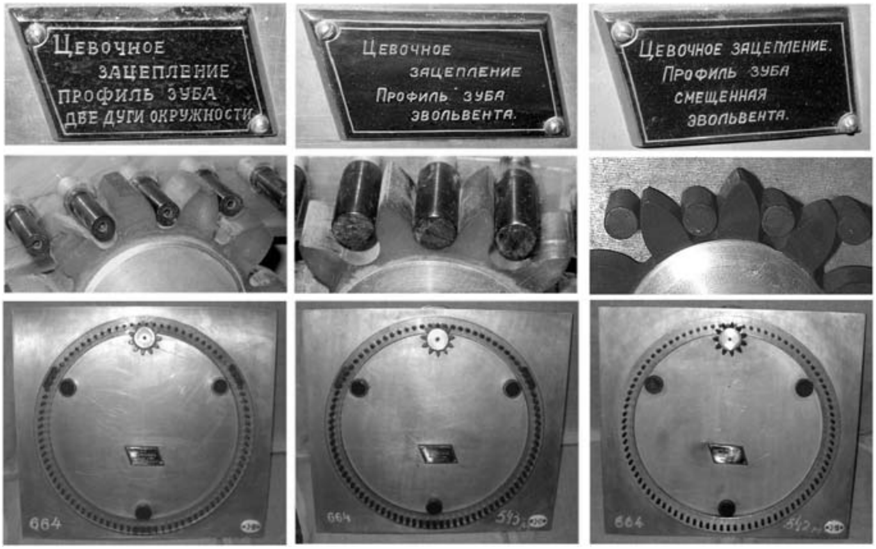


Fig. 3.55: Models of trundle toothed gearing with various profiles of pinion teeth

All three models in Fig. 3.55 have the same constructional execution and the same overall proportions. Trundle teeth number $z_1 = 11$, pin gear number $z_2 = 90$, reduction ratio $u_{12} = 8.18$. J-16 model's pinion tooth profile is made by two conjugate circular arc. J-17 and J-18 models' tooth profiles are involutes of a circle. The tooth of model J-17 is made involutes without displacement of the tool, the tooth of J-18 is made with positive displacement of the tool. Wheel teeth in such gearings are fabricated in the form of an end surface trundle. As the wheels of the models have are large they are made in the form of rings placed on three ball-bearings. Models are provided with the tablets where the various kind of pinion profiles are shown.

Model 381 (Fig. 3.56) is an internal pin gear with a reduction ratio of 2. The pinion has six pin teeth, the wheel has 12 end surface spline teeth. Such a gear is called a "Galloway wheels". In such gearing conjugated wheel profiles are circumscribed by straight lines. The center of the pinion trundle is on the pitch circle r_1. The radius of the pitch circle of the wheel r_2 is two times larger. While circle 1 rolling over circle 2 without sliding the trajectory of any dot of circle 1 in relatively moving wheel 2 will be a straight line. If the dot is replaced by a trundle with radius r_0, then the wheel's

profiles would be parallel lines away from each other for distance $2r_0$. The number on the model does not correspond with the collection classification. It consists only of digits as it was settled during the marking of models in the 1950s of the last century. Probably the model was fabricated between 1950 and 1960 judging by the materials used (duralumin, organic glass and brass).

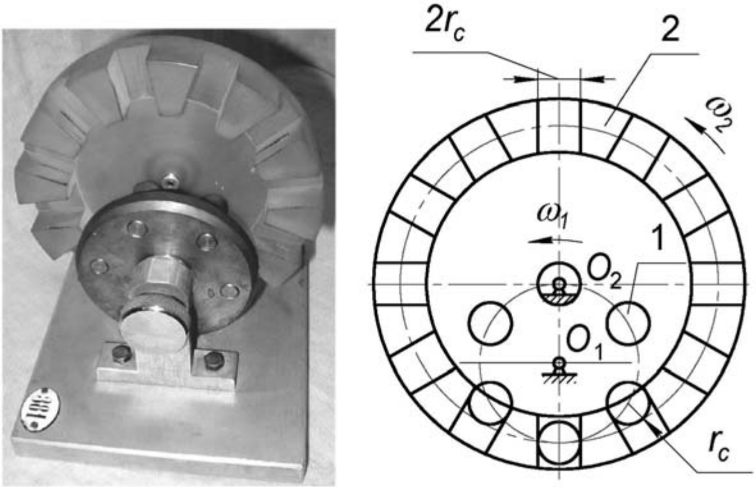


Fig. 3.56: Trundle gearing (Galloway's wheel)

The mechanism of the electric clock (Fig. 3.57) combines the functions of fixation and motion. One can put this mechanism either to a pin gear or to a Geneva mechanism. The input link has two end surface trundles. There are ten figured grooves on the output link. One side of the groove is made along the straight line, another along the circular arc. While one of trundles is meshes with the arc-like groove profile, the motion is not delivered to the wheel. When the trundle approaches the lower part of the cavity and the contact dot approaches the straight part of the groove's profile, the wheel starts its rotation and turns in a one step angle. Rotation finishes when the second trundle approaches the action. Then all phases repeat. The input link makes one turnover while the output makes two. The wheel turns only in one direction. It lacks of such a mechanism.

Approximate epicycloidal toothed gearing is a sort of cycloidal action (Fig. 3.58). In such a mashing the point of the tooth is implemented along epicycloids and the dedendum along the hypocycloid. The radius of the auxiliary circle (centroid) form the hypocycloid of the dedendum and is equal to half of the centroid radius of the wheel. The dedendum in approximate epicycloidal toothed gearing, has a straight profile since the hypocycloid turns into a straight line (Galloway wheels). The model's wheels number of teeth: for pinion – $z_1 = 6$, for wheel – $z_2 = 40$. Toothed wheels module of the model value is m = 15 mm. The model's wheel is made in the form of toothed sector with five teeth, since the diameter of the wheel is large in the case of such module. The model was manufactured in the TMM department during the middle of the last century. This model is similar to models from the Q collection of Reuleaux mechanisms.

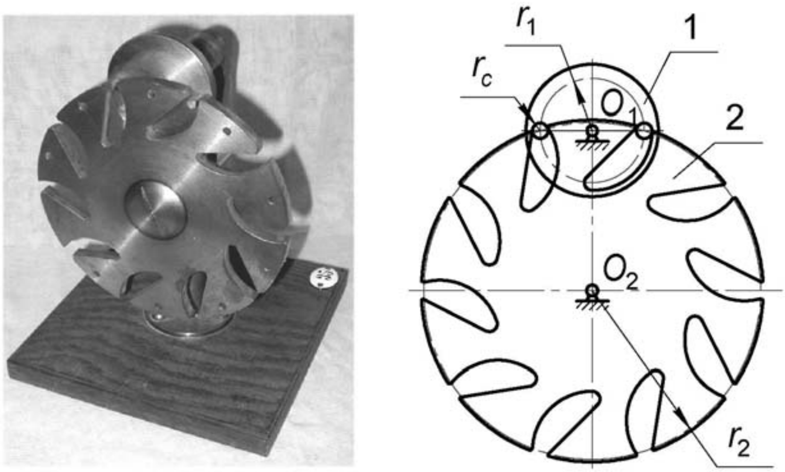


Fig. 3.57: Model of mechanism of electrical clock

Fig. 3.58: Model of cycloidal action

The two models presented in Fig. 3.59 are devoted to pin gears. The number of trundles on the left-hand model's pinion is 2, on rights it is 3. As in the case of model J-21, wheels are fabrication in the form of toothed sectors with six teeth. The wheel teeth's profiles are straight. The models ware fabricated in the TMM department workrooms due to Reshetov's initiative in 1950s and 1960s of last century. They don't have classification numbers.

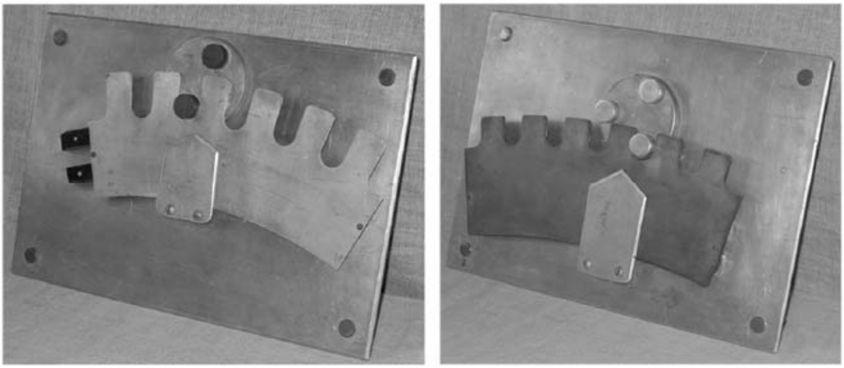


Fig. 3.59: Models of trundle action with a number of trundles on the pinion z_1=2 and z_1=3

3.4.1.1. Gearings with arched teeth

Model of toothed mechanism with arched teeth where the tooth line is fabricated along the cycloid. The model is made by postgraduate student M. Klin following Reshetov's initiative in 1979. The toothed wheels of the model are fabricated from textolite using the technology described in the work [56].

The wheel's teeth are machined with a cutter head so that an approximate quasi-involute action is formed (Fig. 3.60). The cutting motion (the rotation of cutter head B_1 and rotation of blank B_2) and auxiliary motion (the movement of the slide assembly of the cutter head n_3 and additional rotation of blank B_3) are imparted to the blank and the tool to form teeth. To demonstrate the self-adjustment of the pinion, it is connected to the shaft by a cylindrical pair. The shaft can turn over the vertical axis.

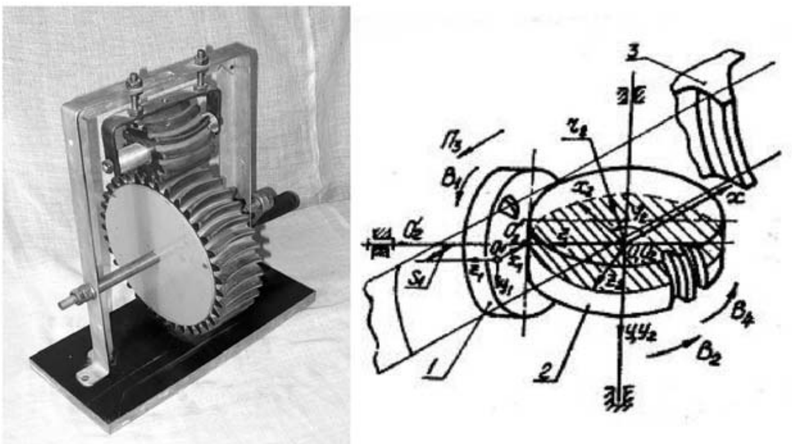
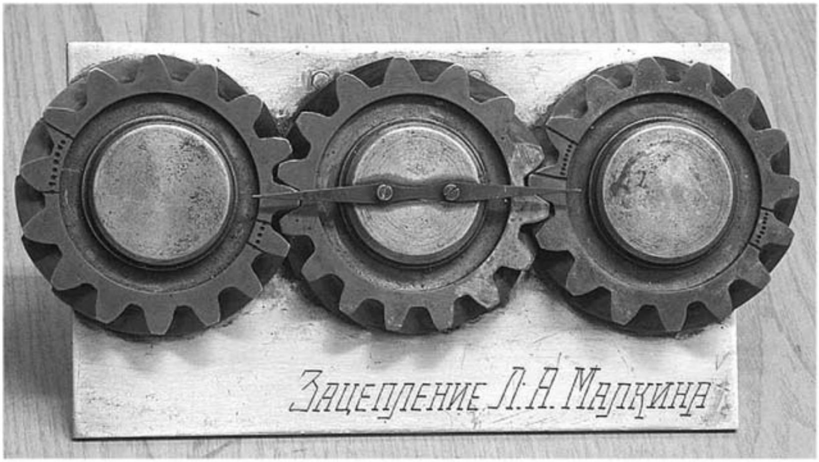


Fig. 3.60: Model of gearing with arc teeth

3.4.1.2. Gearing with Malkin 's toothing [57]

Gearing with Malkin's toothing (Fig. 3.61) is a toothed gearing with variable reduction ratio. It is necessary to remark that the gear-ratio (the ratio of the number of teeth of wheels) is constant in such a mechanism. Teeth in such toothing are round, surfaces of teeth point and profile valley are coaxial and concentric.

The centers of the basic circles are displaced in relation to the wheel rotation axis on value *e*. Therefore, the head-line of contact representing the straight tangent to the basic circles moves in the gear working process. Pitch point *P* at that changes its position on the center line making a linear path – "bipoloid". The reduction ratio



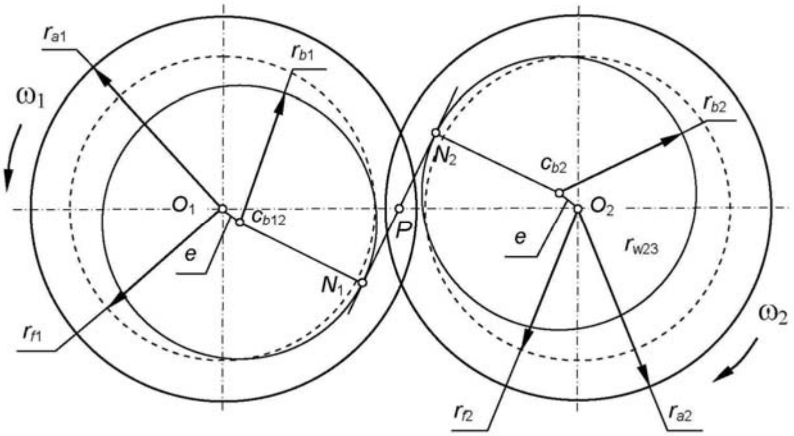


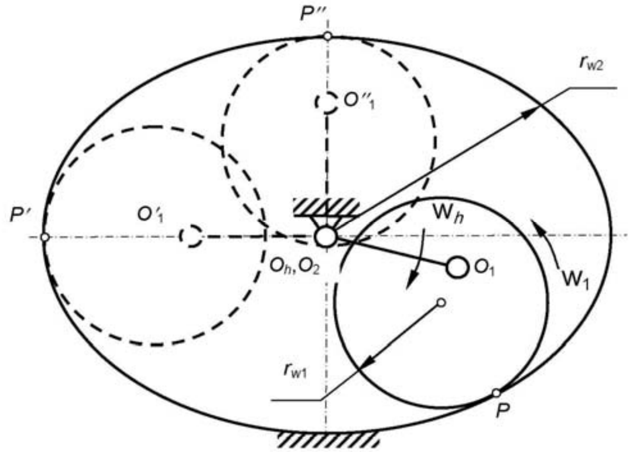
Fig. 3.61: Model of a gearing with circle wheels and a variable reduction ratio (Malkin's action)

changes cyclically. The period of such an oscillation (if the number of teeth is equal) corresponds to one revolution of the driving-wheel. The middle-integral value of the reduction ratio is constant and numerically equal to the gear-ratio.

The amplitude of reduction ratio changing is determined by the amount of eccentricity of the base circle *e.* However, it is impossible to obtain a considerable change of the reduction ratio in gearings with such an action because of the limitation from the rolled kink and cusp (point thinning of teeth). Moreover, teeth in different zones would have different hardness and strength. The model was manufactured by author and presented to the department.

3.4.1.3. The eearines with non-round wheels [58, 59]

Considerable more value of the reduction ratio changing could be received when using non-round toothed wheels (Fig. 3.62).



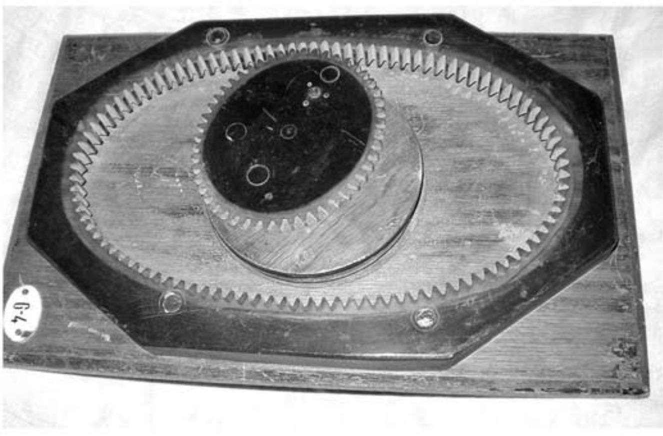


Fig. 3.62: Model of a planetary toothed mechanism with non-circle ram machined wheels

Models of gearings with non-round wheels with external gear are known. Methods of analysis and fabrication of non-round wheels with internal toothing were devised by the TMM department postgraduate student K. Tarkhanov. He projected special devices for the production of such wheels on a gear-shaping machine. With the help of such machines, toothed wheels with external and internal toothing were cut.
At present it is only internal toothing model. The model with external toothing is lost.

3.4.1.4. Novikov's gearing [60]
Gearing with a dotty system of toothing [60] was patented by M. Novikov in 1956. The theory of dotty roundscrewed toothing, formulated in Novikov's doctoral thesis [61], was based on this invention. The theory was based on fundamentally new principles of forming the toothing of cylindrical gearing.
Unlike involute drives designed according to the characteristics of a gear-cutting tool from machine-tool action, Novikov's gearings are designed with the property of strength.
Tooth profiles in face plat front sectional view of this toothing are made by circular arc (Fig. 3.63). Tooth profile of wheel 1 is convex and made by a circular arc with radius ρ_1, tooth profile of wheel 2 is concave and made by a circular arc with radius ρ_2. Radius ρ_2 is bigger than radius ρ_1 an unimportant size.
Usually the radius difference doesn't exceed 5%. When contacting convex profile with concave one the value of reduced radius of curvature in contact point is measured by difference between profile radiuses. The less this value the bigger reduced radius of curvature and according to Hertz theory smaller the contact stress.
Novikov's action has no end surface covering. Lines of action are parallel with wheels axes and normal to drawing plane. Angle of action α_w and length l_{KP} are chosen by designer from recommended range (l_{KP} = 0.05 … 0.2 r_{w1}, α_w = 20°…30°). Novikov gearing has only axled covering therefore it is made with skew tooth with big tooth line tilt angle. Face width is rated by required contact ratio which is recommended to be taken within ε = 1.1…1.2. Complete refusal of end surface covering allowed to reduce dept of tooth and increase bending strength of teeth considerably. Thus Novikov's gearing given other equal conditions in comparison with single-helical involutes gearing provides frequentative increase of strength ratings. Besides dotty contact removes redundant constrains in gearing therefore Novikov's gearing is less sensible to assembling and production errors. Due to higher strength ratings Novikov's gearings must reduce weight and gabarits of reducer greatly and gradually replace involute gearings. Disadvantages of Novikov's gearing are: laborious to make, wide tooth line angle, wide ring gear, changing of reduction ratio when changing axle base.
Novikov created his original system in the middle of 20th century. His creating was accompanied by large-scale state investment in the development and introduction gearings with same principle of action into different branches of industry.
Novikov's idea of action and its great potentials were highly estimated in the scientific world, and the author was awarded the Lenin prize. It is necessary to remark that not every specialist accepted Novikov's ideas at once. Many of them actively counteracted the introduction of this toothing.

M. Novikov

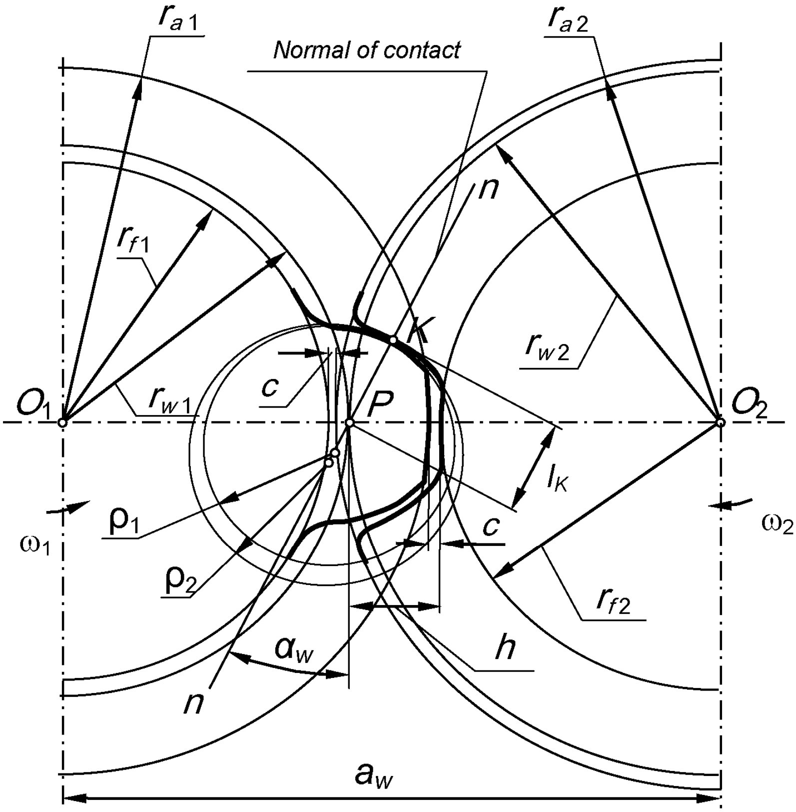


Fig. 3.63: Novikov's action scheme

They considered that it will not be able to compete with involute gearings. Fifty years passed after Novikov's gearing creation and it is time to make a review. Novikov's gearing were introduced in different fields of engineering. Nowadays they are used only in general-purpose reducers. Comparison testing of reducers with Novikov's gearing and involutes gearings show practically identical results in resource and efficiency. The technology of Novikov's gearings production is more complicated. It is difficult to fabricate them with high accuracy. As a result reducers are more expensive and noisy than reducers with involutes action. Today it is early to make final conclusion about future of Novikov's gearing. Practice and testing results refute evident theoretical data.
To find the reasons of it more exact experiments and wide theoretical researching are needed. Such works are carried on [62, 63]. New more hopeful kinds of action are being created on the base of Novikov's ideas.
There are two models of gearing in TMM department collection. Both of this models were made by author and presented to head of the theoretical mechanics department of BMSTU, Professor S. Dobrogursky, and to head of the locomotive engineering department of BMSTU, Professor V. Gavrilenko.
In Fig. 3.64 one can see the model with the tablet: "For much-esteemed professor, doctor of technical science Vladimir Aleksandrovich Gavrilenko from author of new system of action. Novikov. 22.XI.55". On the right side and model with tablet: "For much-esteemed professor, doctor of technical science Sergey Osipovich Dobrogursky from author of new system of action. M. Novikov. 22.XI.55" on the left side. Gearing characteristics in this models are different.

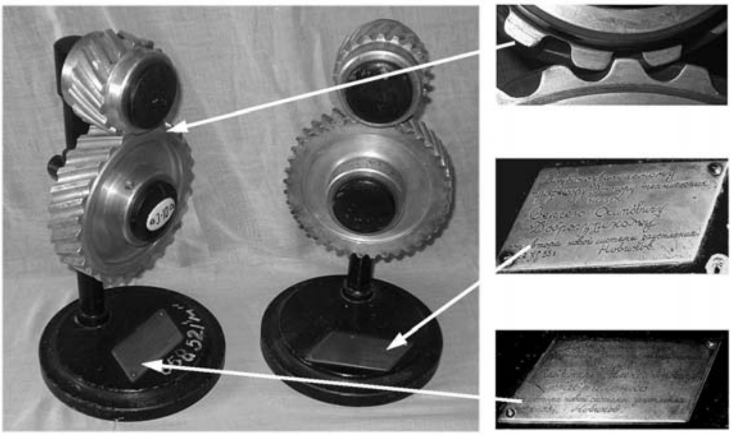


Fig. 3.64: Models of Novikov's gearings presented to Professors Dobrogursky and Gavrilenko by the author

Model J-10 has face width value b = 42 mm, number of teeth: pinion $z_1 = 14$, wheel $z_2 = 28$. Second model has face width value b = 42 mm, number of teeth: pinion $z_1 = 20$, wheel $z_2 = 40$. Both models have same reduction ratio $u_{12} = 2$, diameters of pitch circles $d_{w1} = 90$ mm and $d_{w2} = 180$ mm, axle base $a_w = 135$ mm. The profile of tooth of pinion is convex, and the one of wheel is concave. The direction of helical tooth line on pinion is right, the one on wheel is left. The material of toothed wheel is duralumin D-16.

3.4.2. Compound wheelworks

3.4.2.1. Planetary trains

The section of models of compound wheelworks includes both in-line and planetary trains. Among planetary trains single-row trains are the most commonly used ones. In literature this mechanisms are often called “GM’s mechanisms” after General Motors Corporation, which was the first to make such planetary trains. The single-row planetary train model (Fig. 3.65) was transferred to BMSTU collection from Moscow Institute of Transportation Engineers (in Russian – MIIT) by B. Shitikov. The model has the gear ratio u = 4.8. The teeth numbers of the wheels: central (sun) gear z_1 = 20, satellite gear z_2 = 27, wheels with internal teeth (epicycles) z_3 = 76; the module is: m = 1.5 mm.

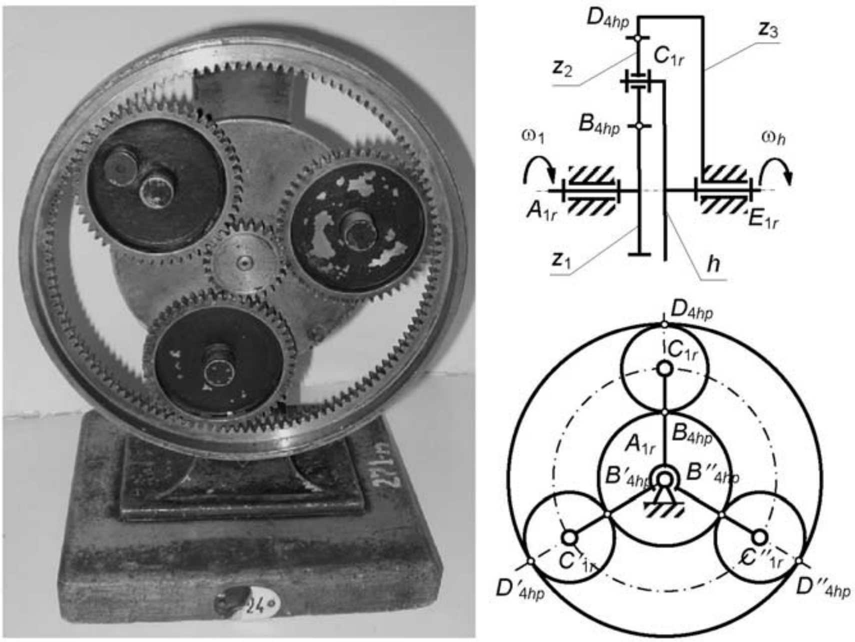


Fig. 3.65: The model of a single-row planetary train with three satellite gears

There are some more models of single-row planetary trains in the collection. Some models are working examples of industrial planetary reduction gears. Others represent models which are results of scientific researches. The models of planetary trains with two and four satellite gears, which photos are shown in Fig. 3.66, belong to the first group. Left model represents the aircraft’s reduction gear with facilitated case details and small width of gear rings of its wheels. The gear ratio of the mechanism u_{1h} = 4.8; numbers of teeth of the wheels: central (sun) z_1 = 37, satellite gears z_2 = 23, epicycles z_3 = 83; the module: m = 1.25 mm. The second mechanism has four satellite gears and

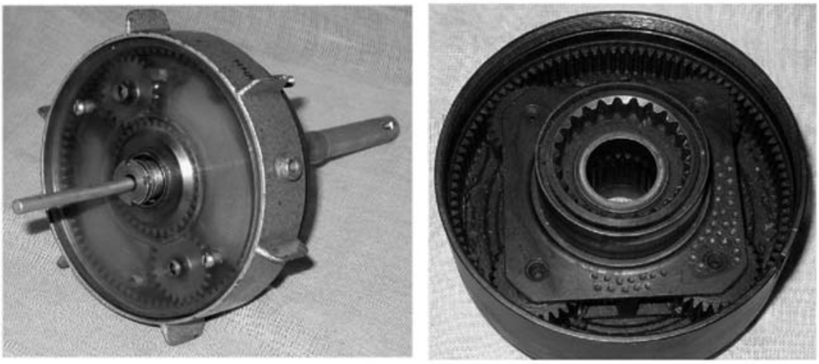


Fig. 3.66: The models of single-row planetary trains made on the basis of workable reduction gears

a normal width of the gear rings of its wheels. The reduction gear has a steel case made integral with the gear ring of wheel z_3. Design philosophy features of the mechanism specify that it is intended for high load transfer. The gear ratio of the mechanism u_{1h} = 3.167; the teeth numbers of the wheels: central (sun) z_1 = 48, satellite gears z_2 = 28, epicycles z_3 = 104; the module m = 1.25 mm.

A series of six models of differential planetary trains was designed and produced at the department in the 1960s. The scheme of a single-row planetary train with one satellite gear is shown in Fig. 3.67. Model N-20 consists of six gears and allows three shafts to rotate simultaneously. It is possible to connect wheel z_3 and planet carrier h to the case

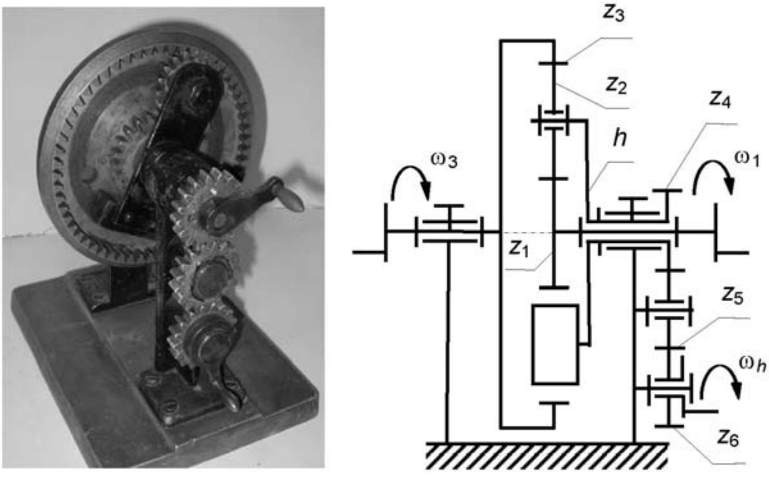


Fig. 3.67: The model of a differential single-row planetary train

by screws and thus to receive mechanisms with various gear ratios. The teeth numbers of the wheels of the model: $z_1 = 23$, $z_2 = 17$, $z_3 = 57$, $z_4 = z_5 = z_6 = 17$. The gear ratios of the mechanism: with stopped planet carrier $u^h_{13} = -2.48$, with stopped wheel z_3 – $u^3_{1h} = 3.48$, with stopped wheel z_1 – $u^1_{3h} = 1.403$. A counterbalance is installed on the planet carrier to balance the satellite gear.

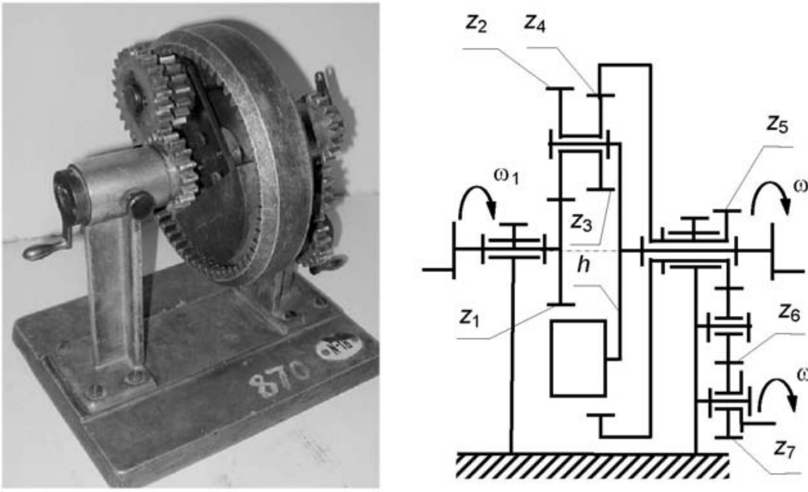


Fig. 3.68: The model of a differential double-row planetary train with one external and one internal gearing

The next model of the series is a double-row planetary train with one external and one internal gearing (Fig. 3.68). Model N-18 has one block of satellite gears which is balanced by a counterbalance installed on its planet carrier. Wheels z_1 and z_4 can be connected to the case by screws and thus we get mechanisms with various gear ratios. The teeth numbers of the wheels of the model $z_2 = 23$, $z_4 = 57$, $z_1 = z_3 = z_5 = z_6 = z_7 = 17$. The gear ratios of the mechanism: with stopped planet carrier $u^h_{41} = 2.22$, with stopped wheel z_4 — $u^4_{1h} = 5.54$, with stopped wheel z_1 — $u^1_{4h} = -0.221$.
Further, we shall examine model N-14 – a double-row planetary train with two external gearings (Fig. 3.69). The model has one block of satellite gears. The satellite gears are balanced by a counterbalance which is installed on link h. It is possible to connect wheel z_3 and planet carrier h to the case by screws and thus to receive mechanisms with various gear ratios. The teeth number of the wheels of the model $z_2 = z_4 = 23$, $z_1 = z_3 = z_5 = z_6 = z_7 = 17$; the module m = 3.5 mm. The gear ratios of the mechanism: with stopped planet carrier $u^h_{41} = 2.22$, with stopped wheel z_4 — $u^4_{1h} = 5.54$, with stopped wheel z_1 — $u^1_{4h} = -0.221$.

Model N-21 (Fig. 3.70) is a double-row mechanism with one external and one internal gearing. Its block of satellite gears, formed by a wheel with external teeth z_2 and a wheel with internal teeth z_3, is the distinctive feature of this model. As well as all other models of the series, this mechanism has one block of satellite gears balanced by a

counterbalance. It has three input links and permits showing various operating modes of the differential mechanism. The teeth number of the wheels of the model: $z_2 = 23$, $z_3 = 57$, $z_1 = z_4 = z_5 = z_6 = z_7 = 17$; the module m = 3.5 mm. The gear ratios of the mechanism: with stopped planet carrier u^h_{41}= -0.403, with stopped wheel z_4 – u^4_{1h}= 1.403, with stopped wheel z_1 – u^1_{4h}= 3.478.

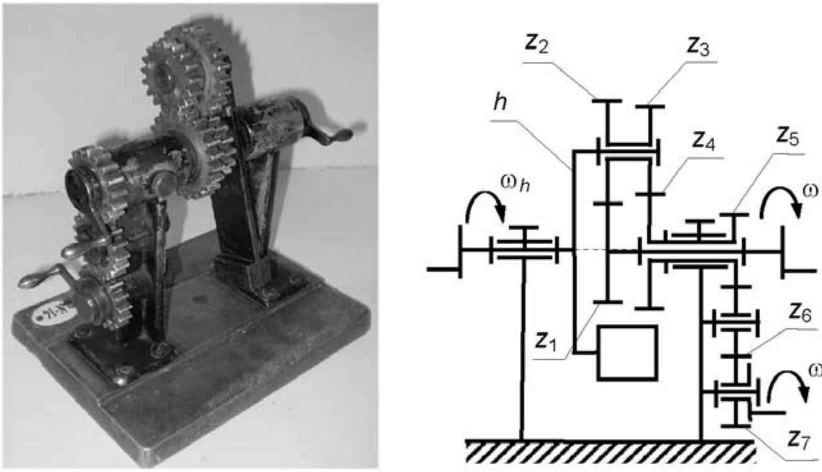


Fig. 3.69: The model of a differential double-row planetary train with two external gearings

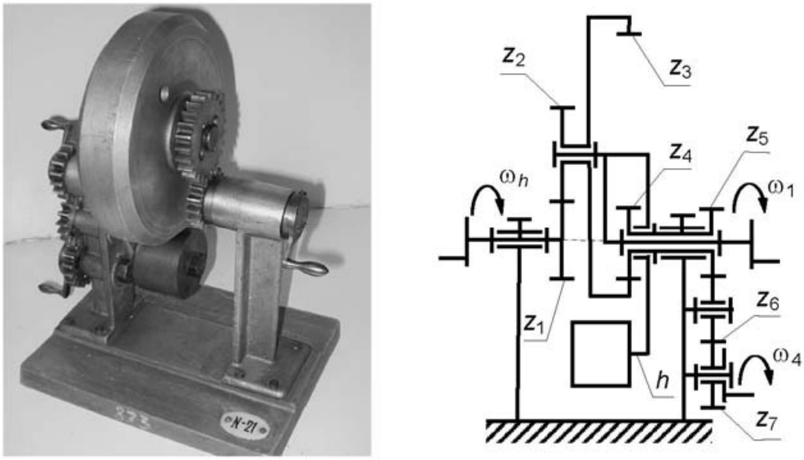


Fig. 3.70: The model of a differential double-row planetary train with one external and one internal gearing

The kinematic scheme of model N-17 differs from the scheme of model N-21 as it has a block of satellite gears formed by two gears with internal gearing (Fig. 3.71). The teeth

numbers of the wheels of the model $z_1 = 20$, $z_2 = 60$, $z_3 = 57$, $z_4 = z_5 = z_6 = z_7 = 17$; the module m = 3.5 mm. The gear ratios of the mechanism: with stopped planet carrier $u^h_{41} = 0.894$, with stopped wheel z_4 – $u^4_{1h} = 9.5$, with stopped wheel z_1 — $u^1_{4h} = -8.5$.

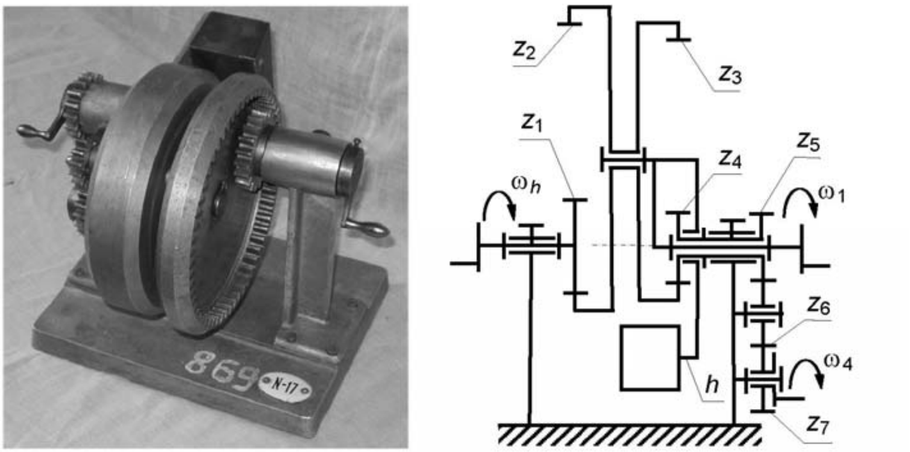


Fig. 3.71: The model of a differential double-row planetary train with two internal gearings

Model N-16, a photo and the scheme of which can be seen in Fig. 3.72, differs from models N-17 and N-21 as it has its block of satellite gears formed by two gears with external gearing. The teeth numbers of the wheels of the model: $z_1 = 20$, $z_2 = 60$, $z_4 = 57$, $z_3 = z_5 = z_6 = z_7 = 17$; the module m = 3.5 mm. The gear ratios of the mechanism: with stopped planet carrier $u^h_{41} = 0.0994$, with stopped wheel z_4 – $u^4_{1h} = -9.06$, with stopped wheel z_1 — $u^1_{4h} = 0.9006$.

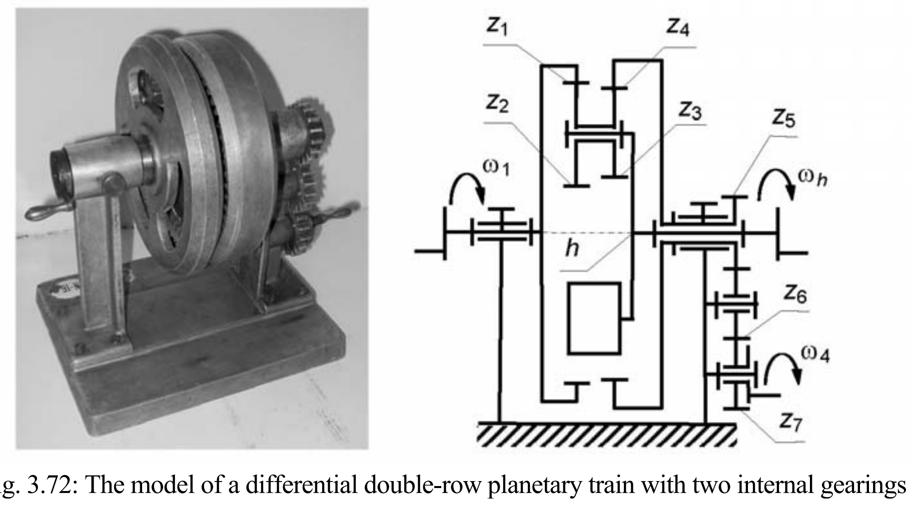


Fig. 3.72: The model of a differential double-row planetary train with two internal gearings

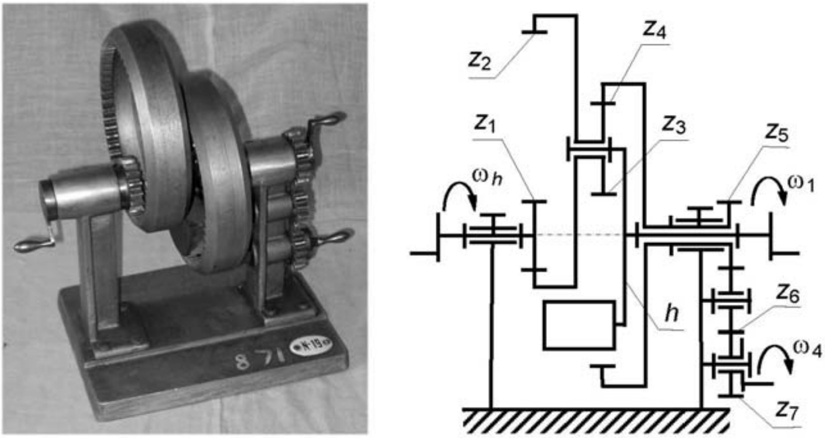


Fig. 3.73: The model of a differential double-row planetary train with two internal gearings

The last model of the series of mechanisms is represented in Fig. 3.73. The model is a double-row planetary train with two internal gearings. The block of satellite gears of this mechanism is formed by a gear with external teeth z_3 and a gear with internal teeth z_2. The teeth number of the wheels of the model: z_1 = 20, z_2 = 60, z_4 = 57, $z_3 = z_5 = z_6 = z_7$ = 17; the module m = 3.5 mm. Gear ratios of the mechanism: with stopped planet carrier $u^h{}_{41}$ = 0.0994, with stopped wheel $z_4 - u^4_{1h}$ = -9.06, with stopped wheel $z_4 - u^4_{1h}$ = -0.9006.

The series of models described includes most of the known elementary (single-row and double-row) schemes of planetary trains. With the help of the series, it is possible to study the kinematics capabilities of differential mechanisms visually at their transformation into a reduction gear by braking of one of their parts.

Further kinematics characteristics of the mechanisms presented are considered. For each gearing of a double-row differential mechanism, it is possible to write down

$$\frac{\omega_2 - \omega_h}{\omega_1 - \omega_h} = \pm\frac{z_1}{z_2}; \qquad \frac{\omega_3 - \omega_h}{\omega_2 - \omega_h} = \pm\frac{z_3}{z_4}.$$

Willis's formula.

$$\frac{\omega_3 - \omega_h}{\omega_2 - \omega_h} = \pm\frac{z_1 \cdot z_3}{z_2 \cdot z_4} = \pm u^h_{31}.$$

Here the gear ratio (ratio of number of teeth) is prefixed with a plus sign when the gearing is internal and a minus sign when it is external.

In this equation the sign before the internal value the gear ratio is positive if both

$$\omega_3 - \omega_h\left(\pm u^h_{31} - 1\right) \pm u^h_{31} \cdot \omega_1 = 0.$$

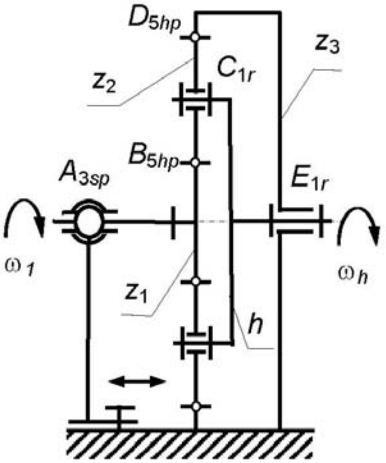


Fig. 3.74: The model of a single-row planetary train with three satellite gears and eliminated redundant constraints

gearings are identical (both internal or both external). If they are different, the sign before the gear ratio u^{h}_{31} is negative.

A number of planetary train models in the collection were made according to schemes developed by L. Reshetov. Redundant constraints are completely or partly eliminated in these models. In model *N-22* (Fig. 3.74) the central wheel shaft is "floating". For this purpose, it is installed into the case in a spherical bearing. The teeth of the satellite

$$q = W_0 + W_l - W^{sp} = 1–0–(6\cdot5–5\cdot4–3\cdot1–1\cdot6) = 1–1 = 0.$$

gears are barrel-shaped, that is why all higher pairs in this mechanism are five-mobile ones. The number of redundant constraints in the mechanism is

The numbers of teeth of the wheels of the model: z_1 = 20, z_2 = 27, z_3 = 76; the module m = 1.5 mm. The gear ratios of the mechanism (when wheel z_3 is stopped) – u^{3}_{1h} = 4.8. For demonstration of angular mobility of the input shaft of the model, the opportunity for the disengagement of wheel *z1* is provided. There too it is necessary to loosen the screws fixing the rack of the bearing and shift the rack in an axial direction.

Model *N-23* (Fig. 3.75) represents a double-reduction planetary train formed by series connection of two single-reduction mechanisms. The first mechanism is practised according to the scheme with a stopped planet carrier, and the second one – according to the scheme with a stopped epicycle (the wheel with internal teeth). The intermediate shaft of the mechanism is self-installing. It is fixed with two higher five-mobile pairs

$$q = W_0 + W_l - W^{sp} = 1–0–(6\cdot9–5\cdot8–2\cdot6–1\cdot2) = 1–0 = 1.$$

in the axial direction. In the radial direction, it is fixed on gearings of wheels z_3 and z_4. The number of redundant constraints in the mechanism

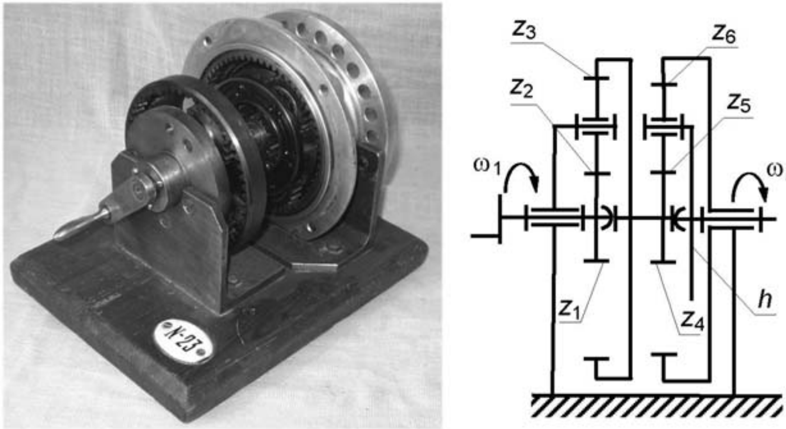


Fig. 3.75: The model of a double-reduction planetary train with three satellite gears and eliminated redundant constraints

The mechanism has only one redundant constraint: it is necessary to hold the axial dimension of the intermediate shaft at a high degree of accuracy.

There are two unusual models of single-reduction planetary trains in the collection. These models show specific capabilities of involute gearing geometry. The model of a single-reduction planetary train with involutes gearing, which has a number of teeth of central wheel with external teeth equal to the number of teeth of the wheel with internal teeth, is represented in Fig. 3.76. The scheme of gearing in this mechanism is also shown in the picture. As wheels z_1 and z_3 are coaxial, their base circles are congruent $r_{b1} = r_{w3}$. Wheel z_3 with internal teeth encloses wheel z_1 with external teeth. Satellite gears are placed in the space between these wheels. To provide this it is necessary to present wheel z_1 with a greater negative displacement, and wheel z_3 – with a greater positive displacement. As possible total displacement is limited by undercutting and teeth sharpening, gears with a minimal number of teeth can be used as satellite gears. For involute gearings with standard parameters of the original profile, this number is equal to five. Displacement at the satellite gear teeth cutting should show are absence of undercutting. However, here the satellite gear teeth come out pointed. There are two involute gearings: an external one – the satellite gear gearing into central wheel z_1 and an internal one – the satellite gear gearing into wheel z_3. The external gearing is negative with the angle of action being equal to 3–5º, the internal one is positive with the angle of action greater than 45º. This model has the number of satellite gears $k = 13$, the teeth numbers of the wheels $z_3 = z_1 = 65$, the teeth number of the satellite gear $z_2 = 5$. The wheels are made of textolite. The front of the model is covered by a transparent plexiglass lid, and the backside – by an opaque textolite lid. There is a handle fixed on the front side of the central wheel for rotation of this wheel. Apparently the model was presented to the department by one of Professor I. Bolotovskiy's postgraduates in about 1973–1975.

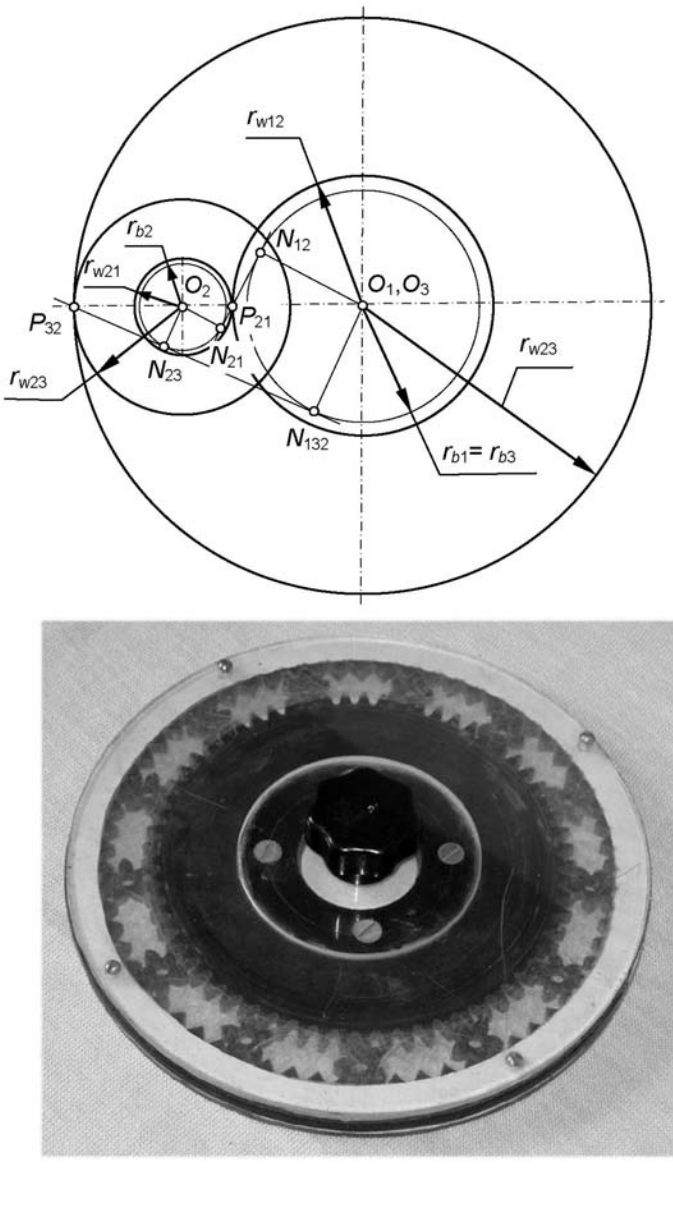
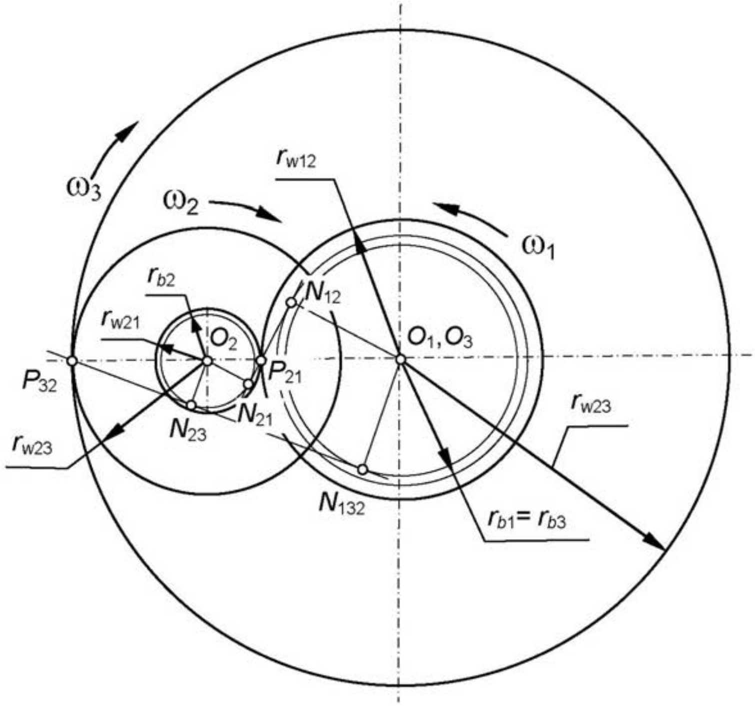


Fig. 3.76: The model of a single-row planetary train with the number of teeth of the wheels $z_1 = z_3 = 65$



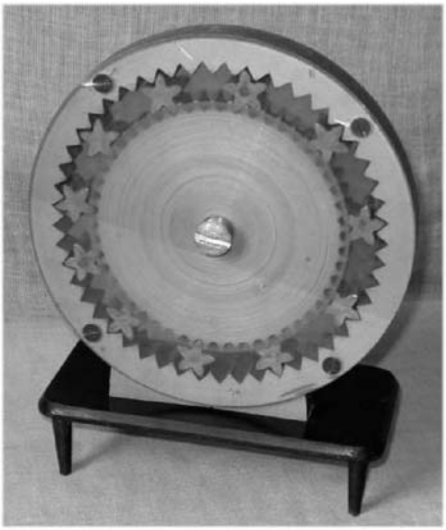


Fig. 3.77: The model of a single-row planetary train with the number of teeth of its central wheel $z_1 = 50$, and of the epicycle wheel $z_3 = 49$

The second, similar model, a photo and the scheme of which are shown in Fig. 3.77, has the number of satellite gears k = 11, number of teeth of the epicycle gear $z_3 = 49$, and number of teeth of its central wheel $z_1 = 50$. In this mechanism the pitch diameter of the wheel with external teeth is greater than the pitch diameter of the wheel with internal teeth. As the teeth numbers of the wheels of the model are less than in the model previously described, it is more difficult to satisfy the boundary conditions for undercutting and sharpening.
Further, the teeth number of wheel z_1 is greater than wheel z_3, therefore the absolute value of displacement on the wheels, cutting should be greater. As the result, the teeth of wheel z_1 have a small undercut, and the teeth of wheel z_3 are pointed. The gears of the model are made of aluminum and its support – of textolite. The origin of the model and its production time are the same as those of the previous model.
A model of Fergusson's planetary train is represented in Fig. 3.78. In this mechanism the teeth number wheel z_1 is equal to wheel z_3 , and wheel z_2 can be optionally chosen. When wheel z_1 is stopped and planet carrier h rotates, wheel z_3 moves translationally. When the model of the mechanism is demonstrated, a glass of water is placed on the support fastened to wheel z_3. The teeth numbers of the wheels of the model: $z_1 = z_3 = 30$, $z_2 = 21$; the module m = 3mm. The model was made in the TMM department workrooms.

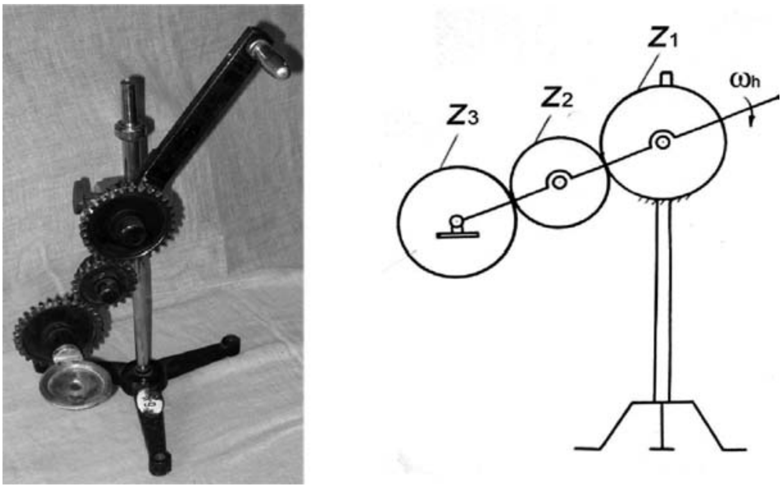


Fig. 3.78: The model of Fergusson's planetary train

Model *H-5* is the most sophisticated of gearing models (Fig. 3.79). The model was made on the basis of a real multi-speed combined reduction gear with a big gear ratio.
The mechanism has two input links, connected to gears z_1 and z_5, and one output link – gear z_{11}. Rotary movement with angular velocity $\grave{u}_1$ imparted to wheel z_1 firstly is transformed in a double-reduction cylinder gear. The reduction gear includes gears

from z_1 up to z_5. Further movement is transferred to three satellite gears z_6, supported by bearings on the hub of wheel z_5. Each satellite gear meshes with two gears z_7 and z_8. Wheel z_8 meshes with three gears z_9. These gears are supported on bearings on fixed axles. Gears z_9 are meshed not only with wheel z_8, but also with wheel z_{10} which is the output link of the mechanism. Wheel z_1 to the output link, the gear ratio of the mechanism, wheel z_7 being stopped, is $u^{(7)}_{1,10} = 406$. Setting the movement of the wheel z_7 is also provided for. Wheel z_7 to the output link gear ratio, wheel z_1 being stopped, is equal to one. The gears of the mechanism have the following teeth numbers $z_1 = 19$, $z_2 = 33$, $z_3 = 21$, $z_4 = 41$, $z_5 = 180$, $z_6 = z_9 = 22$, $z_7 = 78$, $z_8 = 81$ and $z_{10} = 84$. All the gears of the mechanism have the module m = 1.5 mm. Satellites z_6 and z_9 are meshed simultaneously with two coaxial gears which have different numbers of teeth.

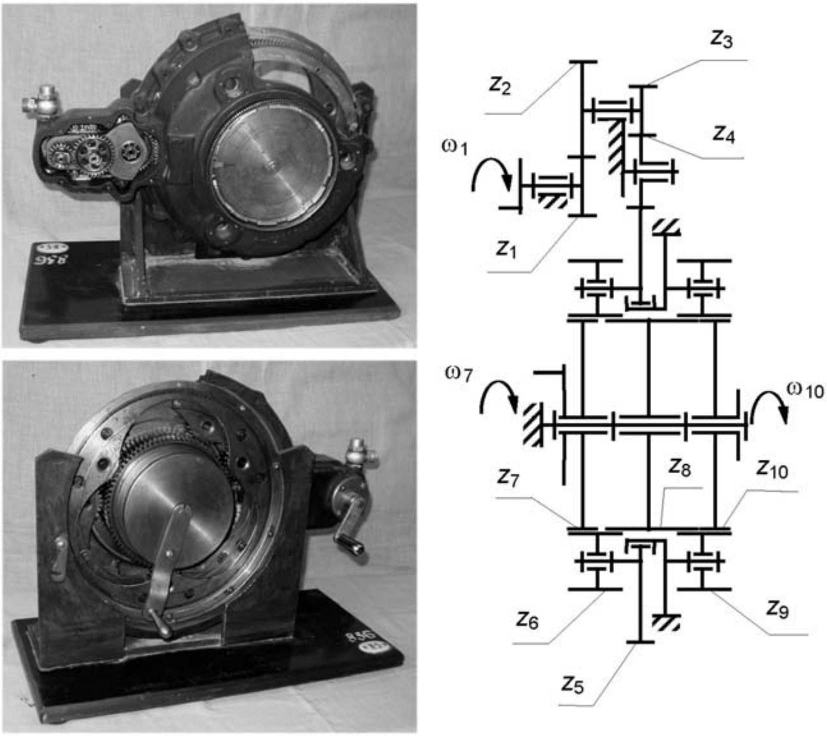
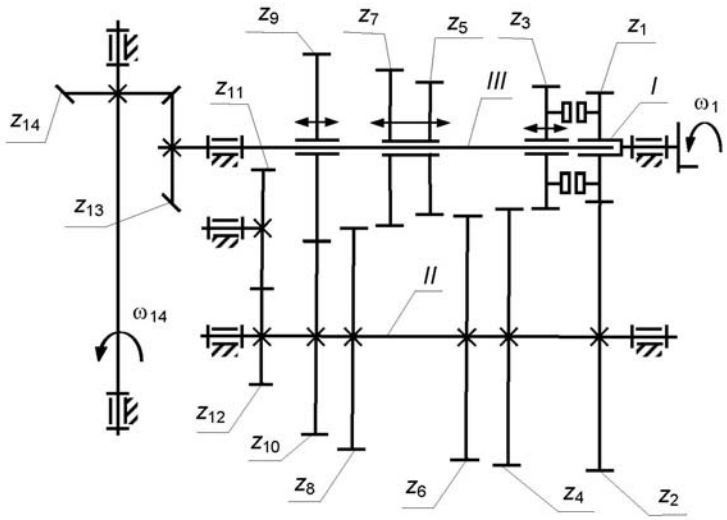


Fig. 3.79. The model of a differential combined planetary train

3.4.2.2. Multi-row gearings

Model H-16 represents the mechanism of a car's five-speed transmission (Fig. 3.80). There are three parallel shafts in the mechanism: I – main drive shaft, II – lay shaft, III – secondary shaft. Pinion z_1= 17 is mounted rigidly on the main drive shaft (input

shaft) and is engaged with gear z_2 constantly. Gears $z_2 = 48$, $z_4 = 41$, $z_6 = 34$, $z_8 = 25$, $z_{10} = 16$ and $z_{12} = 16$ are mounted rigidly on layshaft II. Gears $z_3 = 23$, $z_5 = 30$, $z_7 = 39$ and $z_9 = 48$ are mounted on secondary layshaft III on featherkey. These wheels can move along the shaft axis by a gear-shift device, in that some wheels engage, others disengage. Thus, the gear ratio of the mechanism is changed in steps. Secondary shaft III is connected to the driving axle of the car through the main gear – bevel gear $z_{13} = 24$ and $z_{14} = 41$. There is no bevel gear differential in the model, which provides different speed of rotation of the wheels when the car enters a turn.



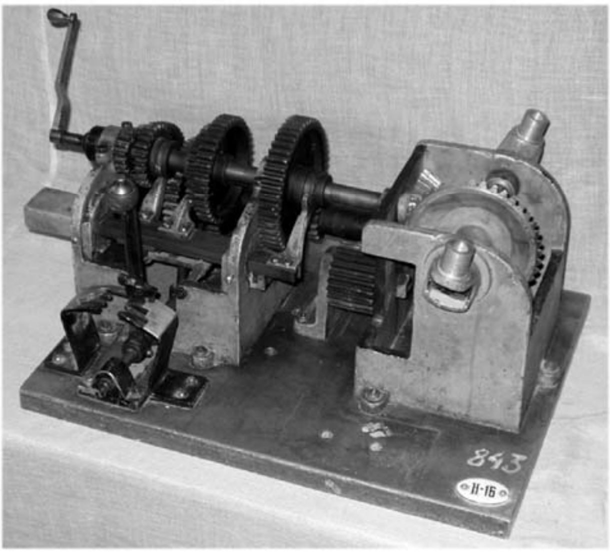


Fig. 3.80. The model of a car's five-speed transmission

When the first gear is in, wheels z_{10} and z_9 engage, which provides the overall gear ratio u_I = 14.45. When upshifting, wheels z_{10} and z_9 disengage and wheels z_7 and z_8 engage. The gear ratio of the second gear is u_{II} = 7.51. When the third gear is in, wheels z_5 and z_6 engage, which provides the gear ratio u_{III} = 4.25. Fourth gear is provided by the engaging of wheels z_3 and z_4. The gear ratio of the transmission in fourth gear is u_{IV} = 2.7. Fifth gear is called a direct gear as it is engaged when shafts I and II clutch by the face gear clutch. The teeth of the clutch are located at the end faces of wheels z_1 and z_3. The clutch is put in by the axial movement of the wheel z_3, all other wheels being disengaged. The gear ratio of the transmission with the fifth gear in is equal to the gear ratio of the main gear u =1.708, which is formed by bevel gears z_{13} and z_{14}. Gears z_{12} = 16, z_{11}= 24 and z_9 form the reverse gear. Getting into reverse gear is carried out by moving of wheel z_9 and engaging it to wheel z_{11}. The model was made in the department using details of a working transmission in about 1950–1960.
The model of variable-speed gear *H*-12 is one of the most interesting models of the collection. The model was created in the TMM department according to the results of postgraduate student Imre Sekey's thesis work [64]. In analytic review of the work a number of similar mechanisms developed in Germany between 1930 and 1960 were described.

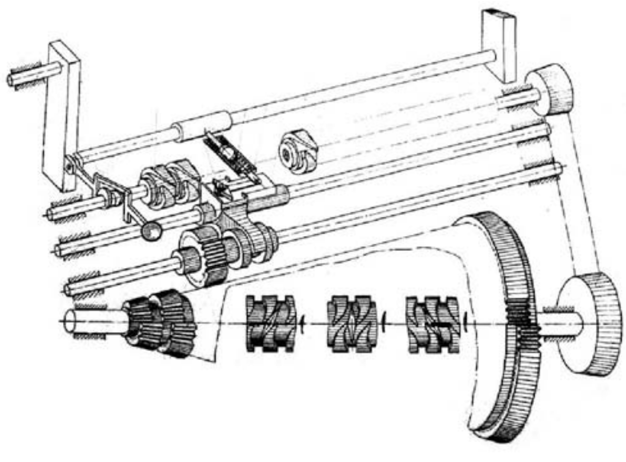
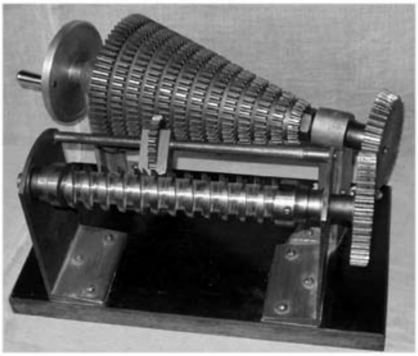


Fig. 3.81: The scheme of a variable-speed gear produced in Germany in the 1950s

The scheme of one of such mechanisms, the design of which is similar to the design of the model being examined, is shown in Fig. 3.81. This gear mechanism possesses the ability to change (switch) the gear ratio without switching off the duty. The scheme of the model and its photo are shown in Fig. 3.82. The main mechanism of the model consists of two gears. One of the wheels z_2 = 26 is an ordinary involute spurgear. The second wheel z_i represents the cone on which external surface 13 gear rings are sequentially placed. Each gear ring has two sectors, and each sector is placed on the 180° arc. The sectors of each ring have half a width of the tooth axial displacement. The teeth numbers of the rings vary from z_{i13} = 68 (at base of the cone) up to z_{i1} = 20

(at cone point) with step $\Delta z_i = 4$. Wheel z_2 moves down the generatrix of wheel z_i cone by means of a reversive screw cam mechanism. Here gear sectors z_i engage with wheel z_2 in turn and transmit movement with a change of the gear ratio under the duty. The gear ratio of the mechanism varies from $u_{2imax} = 2.61$ up to $u_{2imin} = 0.769$. The shaft of screw cam 3 is connected to the shaft of wheel z_i by gearing consisting of wheels $z_3 = 44$ and $z_4 = 44$, with the gear ratio equal to one. All gearings of the mechanism are non-involute gearings or cylinder-conic gearings. In such gearings, conic wheel teeth are cut not by the tool head but by a gear cutter (ram). Here the conic wheel is the spur gear, which has various displacement of the tool (changes under the linear law) in each section on tooth length. The tooth profile surfaces of such a wheel are cylindrical involutes ones, formed not from the base cone, but from the base cylinder. The geometrical theory describing the geometry of gearing in variable-speed gear mechanisms is developed in the work [64]. Similar mechanisms have found application in agricultural and textile machines.



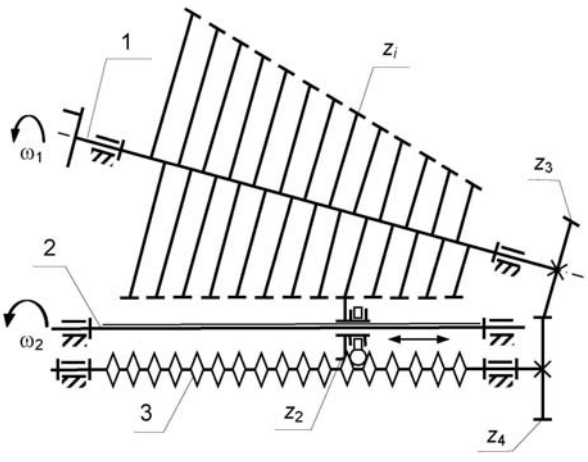


Fig. 3.82: The model of a variable gear ratio wheelwork

3.4.2.3. Crank-planetary trains

Planetary mechanisms on the basis of internal gearing with small difference in the number of wheels' teeth started to be developed in the department at the end of the 1940s. According to the classification of V. Kudrjavtsev [70], these mechanisms belong to the K-H-V type. Crank-planetary mechanism consists of two (one with external and the other with internal teeth), an eccentric planet carrier and a muff. The muff is used to connect the satellite to the output shaft or case. The gear wheel-satellite is installed in the eccentric on the input (high-speed) shaft of the mechanism. When the mechanism works, the satellite realizes plane-parallel motion: its center moves on a circle with a radius equal to the eccentricity and besides the satellite rotates about its axis. It is necessary to transfer only the rotary movement of the satellite to the output shaft. It is carried out by means of muffs of various designs. Mechanisms of parallel cranks with lower or higher pairs, cardan mechanisms, elastic muffs or shafts, gear muffs are used as muffs. The same devices are used in schemes with stopped satellites. In this case, gyration is preserved and rotation about the axis is stopped. The second cog-wheel is fixed to the case or installed on the output shaft.

There are some models of such mechanisms in the BMSTU collection of mechanisms. The first models were created at the end of the 1940s. At that time postgraduate student N. Skvortsova was engaged in the research of such mechanisms. Professor V. Gavrilenko was her academic advisor. In her thesis work [65], she researched the geometry of internal involute gearing with the difference of numbers of wheels teeth equaling one. Recommendations, concerning the choice of geometrical parameters on the basis of the theory of involute gearing were developed in this work. Special attention was paid to research of involutes interference at tooth jam and to constraints on machine gearing: to undercut and cut teeth profiles at processing. Two models created by materials of Skvortsova's thesis are shown in Fig. 3.83.

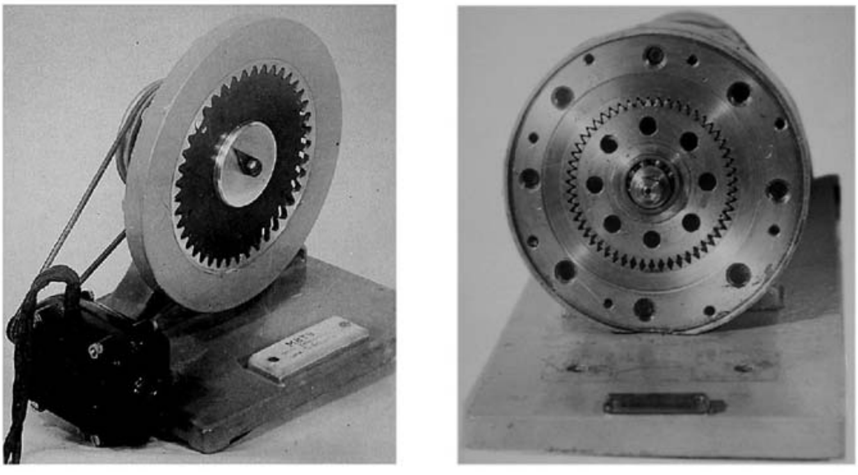


Fig. 3.83: Models of crank-planetary mechanisms developed at the TMM department by materials from the dissertation of N. Skvortsova [65]

The models were manufactured in the department in 1948. The model on the left has been kept till now, the right one unfortunately has been lost. Later in the middle of the 1950s, when research and development work on designing a reducer for a mining combine was performed and one more model of crank-planetary mechanism was designed (Fig. 3.84).

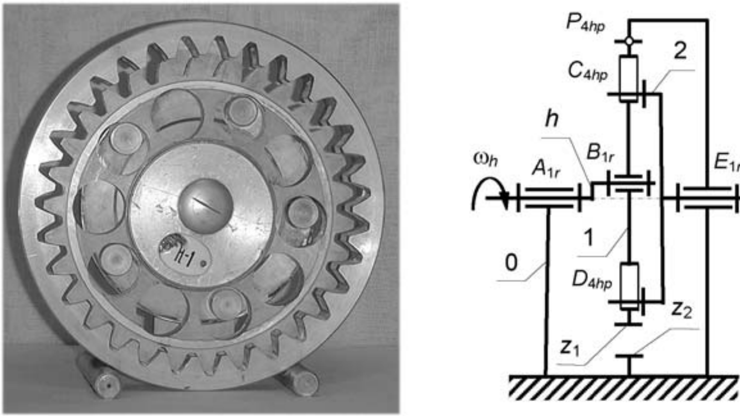


Fig. 3.84: Model of a crank-planetary reducer with a mechanism of parallel cranks with higher pairs as its muff

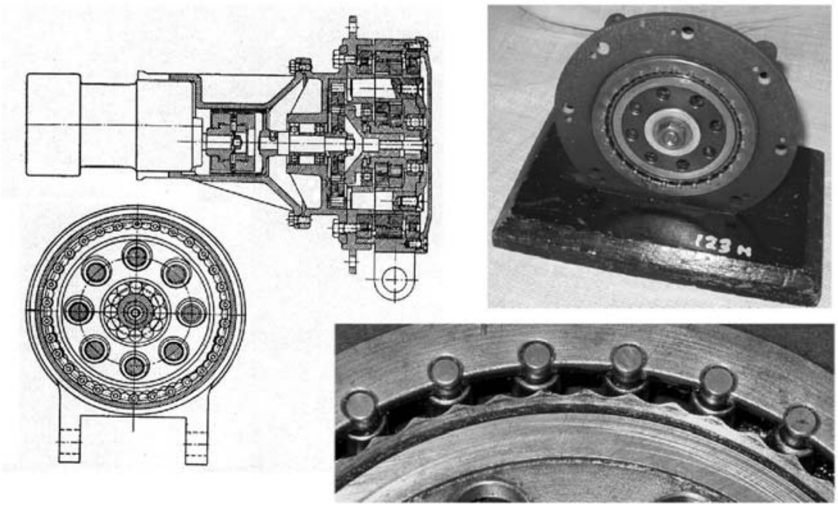


Fig. 3.85: Model of a crank-planetary reducer with trundle gearing (the chassis of a Fokke-Woolf plane reducer)

The model was made in the department's workrooms. In this mechanism, the output shaft is connected to a cog-wheel z_1 by means of the mechanism of parallel cranks with higher pairs. This mechanism can be attributed to trundle gears with internal gearing. Mechanisms of this type have been known since the end of the 19th century. The collection of mechanisms of F. Reuleaux has a model of such a muff and the muff's description is given in F. Orlov's book [11]. The first mechanisms of the K-H-V type had no involutes but cycloidal or trundle gearing [66, 67]. There are two models of such a mechanism in the collection. The first model is made from a reducer of the chassis's rise mechanism of a Fokke-Woolf plane (Fig. 3.85).

The mechanism consists of a bin wheel with internal teeth and a trundle with external cycloidal profile teeth (Fig. 3.86). The mechanism has two wheels with external teeth which are installed on the cage in an antiphase, i.e. eccentricities of its installation are opposite in direction. The number of teeth of these wheels are identical and are z_1= 53. The wheel with internal teeth has only 27 trundles, i.e. everyone, even the trundle is missed. The designed number of trundles of this wheel, which defines gear ratio of the mechanism, is z_2 = 54. Movement is transferred to the input shaft from wheels z_1 through a trundle muff with internal gearing. The number of trundles in the muff is eight. The gear ratio of the mechanism when wheel z_2 is stopped is $u^2_{h1} = z_1 = 53$.

Fig. 3.86: Planetary mechanism of a hydromotor with trundle gearing

The working pair of the planetary hydromotor is the basis of the second model with internal trundle gearing. In 1969, postgraduate student B. Shman was engaged in the development and research of such mechanisms. Professor V. Gavrilenko was his academic advisor. Work was carried out in collaboration with the Omsk factory of hydraulic machines. The working pair used in the model is one of experimental samples. The number of teeth of the rotor is z_1= 6, the number of stator trundles is z_2 = 7. The model was made by Ass. Professor of the TMM department, V. Tarabarin in 2004.

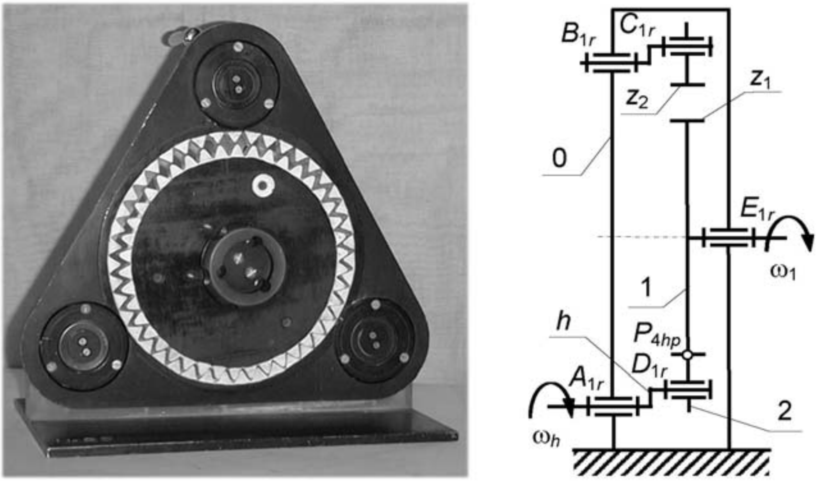


Fig. 3.87: Model of crank-planetary reducer with mechanism of parallel cranks with lower pairs as its muff

Some models of crank-planetary mechanisms and mechanisms of gear muffs were presented to the department by postgraduate student B. Tsilevich. In 1968 the research section of the TMM department was engaged in designing and researching a crank-planetary reducer to drive a radar antenna. The mechanism of the reducer was made in according with the scheme in which the satellite with internal teeth had a circular path of movement by means of the mechanism of parallel cranks with lower pairs. The number of cranks in the mechanism was three. The reducer was designed under the direction of B. Tsilevich by, an engineer of the department, V. Tarabarin. In 1969, the reducer was manufactured and tested by the factory [68]. During tests the efficiency of the reducer was defined and also its vibration activity was estimated. Resonant modes which were explained by low accuracy of the designed balancing of the satellite of the mechanism, were revealed in the working range of frequencies. Having defended his thesis in 1974, B. Tsilevich designed and manufactured the model of planetary reducer H-25 carried out according to the same kinematics scheme. A photo of the model and its scheme are shown in Fig. 3.87. The model of the crank-planetary mechanism with gear muff H-26 and model of gear muff H-27 were made simultaneously. In his thesis work [69], B. Tsilevich was engaged in researching the geometry of internal involute gearing with small difference of wheel in the teeth and the geometry of a gear muff. The models created by him were real corroboration between the efficiency of the designed methods developed in the thesis and recommendations on designing. A photo of the model of the gear muff and the scheme of its gearing is shown in Fig. 3.88. The numbers of wheels' teeth of the muff are identical $z_1 = z_2 = 33$. When the coefficients of displacement of the original profile on the muff's wheels are equal, they form a congruent pair among themselves, i.e. the wheel with external teeth fills the internal space of the wheel with internal teeth as molten metal fills the casting form.

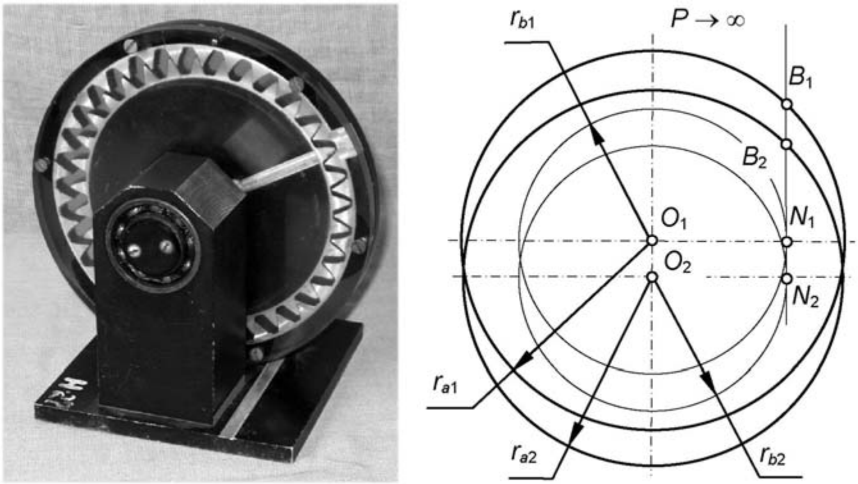


Fig. 3.88: Model of a gear muff with internal involutes gearing

This comparison isn't absolutely exact because there is a radial backlash between the points and clearance of teeth.

The presence of the muff's axle base is possible only due to the difference of displacements. Displacement of the initial contour of the wheel z_2 should be more than a displacement of the wheel z_1. The greater the axle base provided in the muff, the more the difference of displacements of wheels should be. As the displacement increases, the wheel's teeth become pointed and the engagement factor decreases. Therefore, gear muffs can be used when there are small distances between the axes of wheels. The gearing of the muff is a special kind of involute gearing. The basic circles of wheels are identical. The contact normal line to profiles, which is a tangent to the basic circles in involute gearing, is parallel to the line of the axle base and doesn't cross it. Thus, the pole of gearing is in infinity and initial circles have infinite semidiameters. The four-bar mechanism in which the joint pins are located in points O_1, O_2, N_1 and N_2 is the replacing linkage of the gear muff for all possible positions of the links. Angular velocities of cranks O_1N_1 and O_2N_2 of this mechanism are identical and coupler N_1N_2 makes a translator motion on a circular path with radius $O_1N_1 = O_2N_2$. The angle of action of the muff is $a_w = n/2$.

Due to the restrictions specified, gear muffs make good use in crank-planetary mechanisms with internal gearing with a difference of teeth numbers $z_d = 1$.The model of such a mechanism is shown in Fig. 3.89. In the mechanism, the wheel with internal teeth z_2 is fixed in case 0 and gear rings z_1 and z_3 have an identical number of teeth and identical geometrical parameters. Link 1 is installed on the bearing B on crank shaft *h*. Rotary movement from link 1 to output link 2 is transferred by the gear muff which is formed by cog-wheels z_3 and z_4. In internal gearing transmission, the number of teeth are $z_1 = 42$, $z_2 = 43$, the module of gearing is $m = 4$ mm. In the gear muff, the number of teeth are $z_3 = z_4 = 33$, the module is $m = 5$ mm.

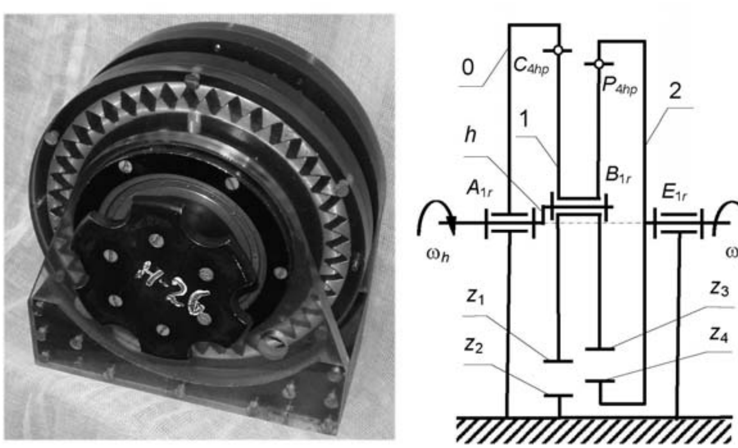


Fig. 3.89: Model of a crank-planetary mechanism with a gear muff with internal involutes gearing

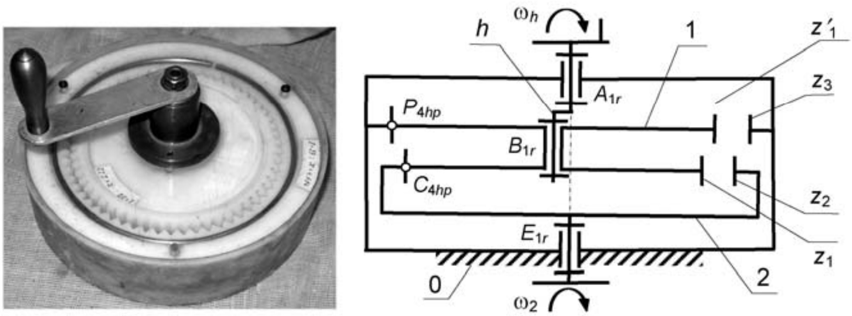


Fig. 3.90: Model of a crank-planetary mechanism with a gear muff with internal involute gearing (Izhevsk Mechanical Institute)

The model of the similar gear mechanism (Fig. 3.90) was probably created by the Ass. Professor N. Voronov of Izhevsk Mechanical Institute. The scheme of this mechanism doesn't differ very much from the scheme of model *H-26.* The numbers of teeth of wheels in the transmission are $z_1 = 59$, $z_2 = 60$, the module of gearing is $m = 2$ mm. In the gear muff, the number of teeth of wheels are $z_1 = z_3 = 59$, the module is $m = 2$ mm. However, mechanisms differ essentially in the geometry of gearing. In mechanism *N*-26, the number of teeth in transmission is considerably (approximately in one and a half times) less. To provide the necessary front engagement factor in the gear muff, the module of gearing in it was increased from $m = 4$ mm to $m = 5$ mm. The diameters at the middle of the teeth height of wheels z_1 and z_3 should be approximately identical, therefore the number of teeth on the wheels of the muff was reduced to $z_3 =$

z_4 = 33. In Voronov's model at number of teeth z_1 = 59, the engagement factor in the muff is provided without an increase in the module. Therefore, the satellite has one gear ring, which meshes both with wheels z_2 and z_3. The teeth height of wheel z_1 is identical at both mesh zones with wheels z_2 and z_3. In such mechanisms, the tooth height in the muff often increases approximately up to the height of the tool tooth for an increase in the engagement factor. This is possible because the muff tooth doesn't fall outside the limits of the hollow of the interfaced cog-wheel. In the model factors of displacement of the tool used for cutting the wheels x_1 = 2.22, x_2 = 2.9 and x_3 = 4.684. Cog-wheels and the case of the model are made from fluoroplastic and the model is covered by transparent plexiglass lids at its endfaces.
In the mechanisms described above, internal gearing transmissions can be used at both stages of the mechanisms. In this case, the gear ratio of the mechanism considerably increases and its size reaches several thousands. However, in this connection, the efficiency of the mechanism decreases appreciably. In the collection, there are no double-reduction planetary mechanisms with cylindrical cog-wheels.
Model H-9 is double-reduction planetary mechanism with cone gears. A detailed description of such mechanisms can be found in the book by B. Pavlov [70]. Here the type schemes of mechanisms and formulas for calculation of the gear ratio are shown, descriptions of various designs of mechanisms, results of their tests, and data on the accuracy and efficiency of gears are given.

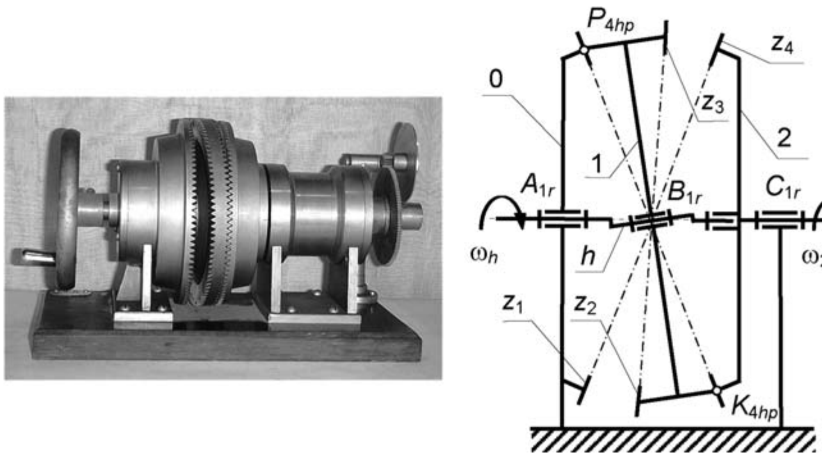


Fig. 3.91: Model of a two-stage crank-planetary reducer with conic gearing

A photo of this model and its scheme are shown in Fig. 3.91. The number of teeth of the wheels of the model are z_1 = 99, z_2 = 100, z_3 = 101, z_4 = 100. The general gear ratio of the mechanism is u^{1}_{h4} = 10,000, i.e. at the turn of the input shaft at 100 revolutions, the output one will turn 0.01 revolution. Rotation of the output shaft is practically imperceptible. Therefore, an increase gear with a gear ratio u = 10, which connects the output shaft to an arrow-index, is established on the slow-speed shaft.

Satellite 1 is established on the input shaft on bearing B. The axis of the satellite and the input shaft axis intersect at an acute angle.
As input shaft h rotates, the satellite precesses with kinematic pair C as its center. The gear ring of the satellite z_2 revolves around gear ring z_1 and gear ring z_3 revolves around gear ring z_4. The number of teeth on wheels z_1, z_2 and z_3, z_4 differ by one, therefore for one revolution of the input shaft, one of the wheels shifts by one angular step with respect to another. The model was manufactured at the TMM department during the middle of the last century.

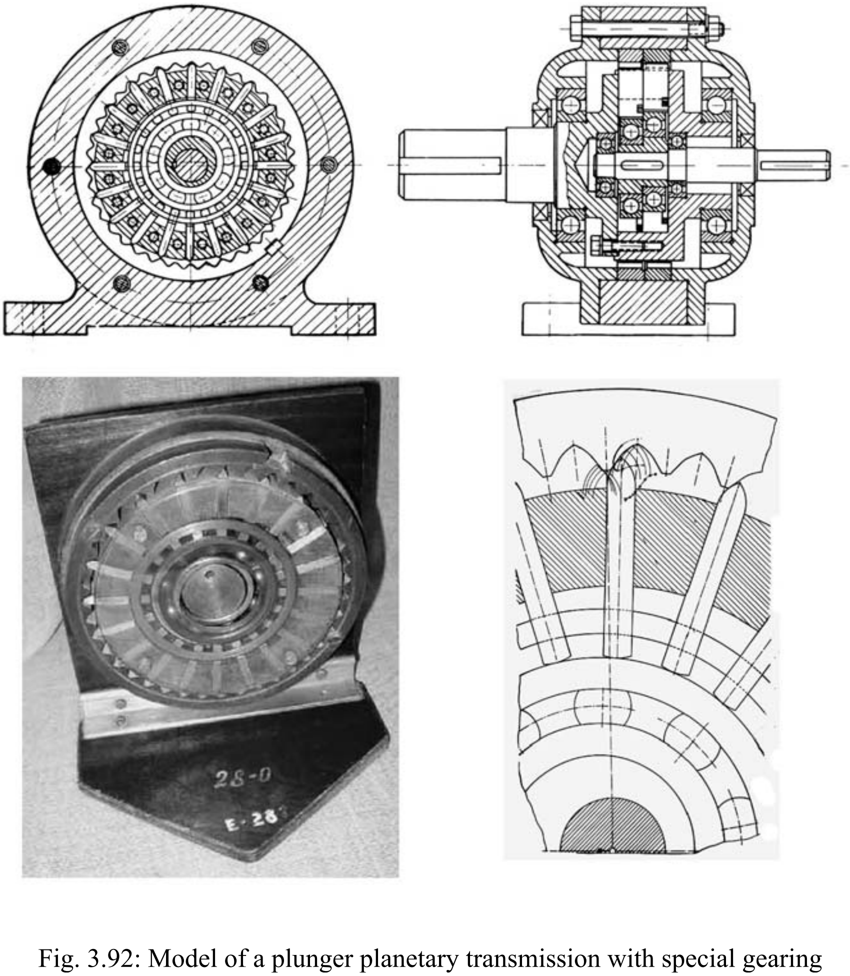


Fig. 3.92: Model of a plunger planetary transmission with special gearing

The next mechanism, *E*-28, was described by M.W. Pitter in 1925 [67] and called a heliocentric reductor. The mechanism has a special geometry of gearing that provides uniform rotation of the output shaft. The mechanism was applied as a reducer in drives of presses, conveyors, fans, machine tools and marine plants. The mechanism consists of an input shaft with the ball-bearing established eccentrically on it, a separator, plungers and wheels with internal teeth. The plungers are located in grooves of the separator and can move in them in a radial direction. The bottom end of the plunger has a cylindrical surface, the top is executed in the form of a tooth with a special side surface profile. In the assembled mechanism, the bottom end of the plunger contacts the external surface of the external ring of the bearing and the top meshes with the teeth of the cog-wheel. As the input shaft rotates, the plungers move in the grooves of the separator. The teeth of the plungers mesh with the teeth of the wheels and drive the separator in rotation. The cog-wheel of the model is fixed inside the case. The mechanism, the design of which is shown in Fig. 3.92, also has the same scheme. The model of the mechanism was made in the TMM department, details of which were fabricated by V. Yastrebov. The scheme of the gearing and the drawing of the reducer were taken from a draft in V. Jastrebov's article kept in the V. Gavrilenko archives. The article was sent to the editors of "Mechanical Engineering Herald" magazine by the author. M. Groman and V. Gavrilenko gave negative reports and it was returned to the author for revision. The geometry and kinematics of the given gear mechanism were considered in the article. It offered to profile of the plungers' teeth on the arc of a circle. Photos of the details, from which model *E*-28 was fabricated, were shown in the article.

Harmonic drives

The American inventor C.W. Musser took out the first patent for a harmonic drive in 1959 [71]. A little later he patented some more constructive modifications of such a gear mechanism. The rights to the basic patent and some of the inventions connected with it were owned by a shoe machines manufacturing company (United Shoe Machinery). C.W. Musser was an adviser to this company for some time. The company produced a series of harmonic drive reducers with various parameters of a flexible shell and various parameters of gearing. Service life tests of the reducers were carried out. According to the results of these tests, the parameters of a flexible shell and the geometry of gearing, which provided necessary durability of the drive, were chosen. These parameters were assumed as a basis for a standard row of harmonic drive reducers.

Harmonic drives possess a number of advantages: large gear ratio in one stage (u = 60 … 300), multiple contact and multizone gearing, high load-carrying capacity (and therefore, small dimensions), small backlash and moment of inertia of the mechanism, and the opportunity of movement transfer through a hermetic wall.

In the Buman University Ass. Professor Ya. Semin was the first researcher in this field. H. Kazykhanov, a postgraduate student of the TMM department, studied the geometry of wave gearing in his thesis. In 1970 he was training in the USA where he visited C.W. Musser and stayed at Musser's for some days. He found out that Musser's first harmonic drive was made in his home workrooms. As he didn't have gear-cutting tools, he made teeth of both flexible and rigid wheels on a lathe with a triangular section cutter. The same triangular teeth were shown in pictures of the patent. This

information was unknown to the majority of harmonic drive geometry researchers. Debates on the form of teeth profiles of a harmonic drive were held for a long time, several researchers considering that the teeth should be triangular with rectilinear sections. For professional inventor, Musser's the harmonic drive was one of the main episodes of his activities. In 1970, he was dealing with problems of spectral analysis and hardly recollected how he had created harmonic drives.

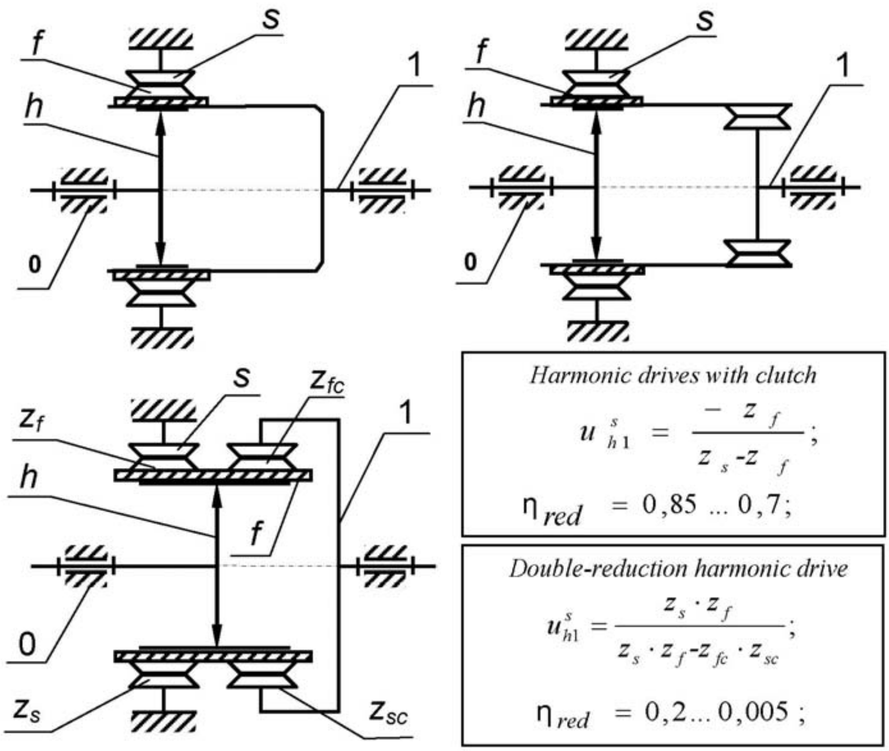


Fig. 3.93: Typical scheme of harmonic drives with cylindrical cogwheels

Harmonic drives opened up a new section in the theory of gearings. D. Dudley, in the reference book [72], considers the formation of new ideology of designing, which he suggests naming "elastokinetic". Nowadays, the majority of experts consider harmonic drives as a variety of planetary trains. Some researchers relate crank-planetary mechanisms to harmonic drives, naming them "harmonic drives with rigid parts" [70]. In a crank-planetary mechanism with a stopped satellite, points of the satellite move along circles of an eccentricity radius. In harmonic drives with a stopped flexible shell, points of the flexible shell move along closed elliptical curves the major semiaxis of which is equal to the sum of the radial deformations of the flexible shell on minor and major axes. In the first case, the trajectory of points are trajectories of movement of the rigid part - the satellite. In the second case, trajectories are travel that are provided by

the deformation of the gear ring of the flexible shell by the wave generator. Nowadays, there is no settled definition of a harmonic drive. Authors adhere to the following formulation of this definition: “A harmonic drive is a planetary train in which transfer and transformation of movement is carried out as the result of relative travelling of the teeth of its wheels through the cyclic deformation of the wave generator”. Dudley [2] suggested considering a harmonic drive as two sequentially connected mechanisms: the harmonic drive itself and a clutch which connects a flexible gear ring with the output shaft or the case. There are various designs of clutches (Fig. 3.93): flexible glasses, flexible pipes with grooves on their end, and harmonic drive gear clutches. The last ones are harmonic drives with the number of teeth of the flexible shell equal to the number of teeth of the circular spline. Harmonic drives are represented in the collection of mechanisms of the TMM department by several models and several full-scale models.

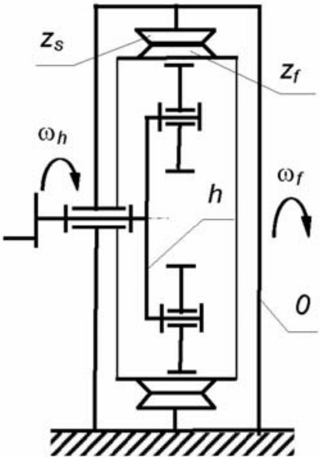


Fig. 3.94: The harmonic drive with a flexible-ring and a double-roller wave generator

In 1970, the TMM department decided to include a harmonic drive section in its lectures. Two models were made for the demonstration of harmonic drives during lectures. The first model (Fig. 3.94) is a flat demonstration model of a harmonic drive with a roller wave generator. The model was designed by the engineer T. Komarova in 1970. It consists of a rigid gear fixed to a frame, a double-roller wave generator and a flexible gear. The drive has no output shaft and clutch. Input movement is set to the shaft of the wave generator and the output movement is the rotation of the flexible gear. The number of teeth of the rigid gear $z_s = 152$, the flexible gear $z_f = 150$, the gear ratio $u^s_{hf} = 75$. When the model was being manufactured, some spare sets of gears were cut. However, the model interested members of the Refresher Courses for Technical University Teachers. Many of them wished to own such models. All spare sets of gears and schemes of the model were given to the members of the Courses. Therefore, when the flexible gear of the model cracked at the tooth hollow, there was nothing to replace it. Nowadays, restoration of the model is needed - it is necessary to fabricate a new flexible gear.

The second demonstration model of a harmonic drive (Fig. 3.95) was manufactured in 1970 from details of the reducer designed in 1967 by a postgraduate of the department, H. Kazykhanov. The model is a mechanism with a flexible gear of glass. The wave generator of the model is a double-roller one. To make all the elements of the drive visible, the model has a plexiglass case. The number of teeth of the rigid gear $z_s = 202$, flexible gear $z_f = 200$, wave generator shaft-to-flexible-gear shaft gear ratio at a stopped rigid gear $u^s_{hf} = 100$. Both models were made in the TMM department.

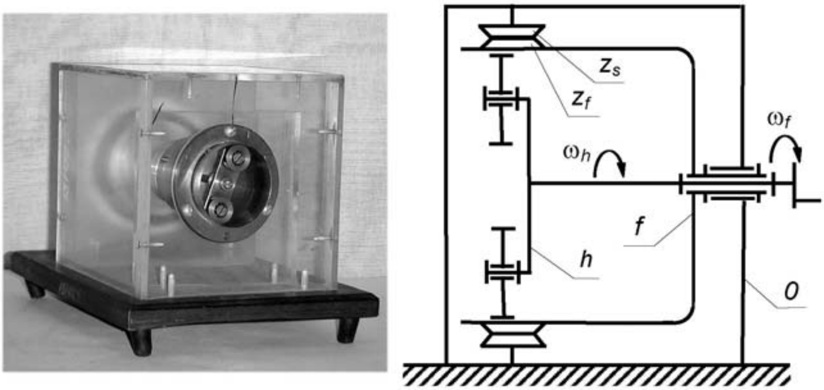


Fig. 3.95: The harmonic drive with a flexible-glass and a double-roller wave generator

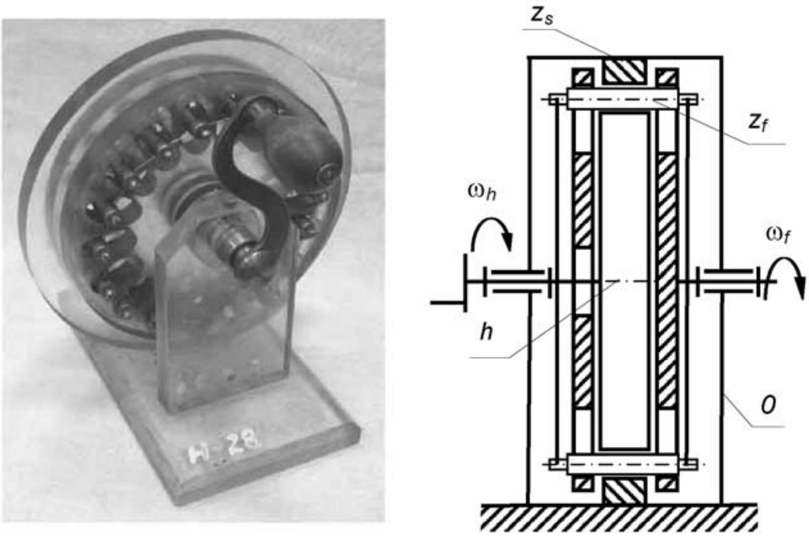


Fig. 3.96: The harmonic drive with a flexspline-chain and a cam wave generator

Another unusual model of a harmonic drive was created by H. Kazykhanov in the first year of his postgraduate course in 1966. While studying material a bout harmonic drives, Kazykhanov found a description of the mechanism in a foreign magazine. The teaching foreman of the department, L. Serebryakov, made a model according to this description. The input shaft of the model (Fig. 3.96) is connected with a double-wave cam wave generator. The flexible gear of the mechanism is performed as eight rollers-trundles, which are connected between themselves by two flexible wire rings. The rollers are fixed to be motionless with an even pitch on the rings and form an elastic chain. The generator is located inside this chain so that the rollers of the chain roll over the external surface of the wave generator cam. The rollers of the flexible gear form the internal gearing with the cycloid teeth of the rigid gear. The number of teeth of the rigid gear $z_s = 16$, the trundle number of the flexible gear $z_f = 18$, wave generator shaft-to-flexible-gear shaft gear ratio at a stopped rigid gear $u^s{}_{hf} = 8$. The movement from the trundles to the output shaft is transmitted through a clutch. The clutch is formed by two disks fixed on the output shaft. Sixteen profiled apertures that settle down the trundles are cut in these disks. Such mechanisms can use multi-row spigot-roller chains as flexible gears. As chain deformation by a generator is not connected with its durability, such harmonic drives can provide a very large range of gear ratios — from one up to several hundreds. The disadvantages of such drives are difficulties appearing when the clutch profiled port and cycloid teeth of the circular spline are manufactured. There is an application of such mechanisms in a power marine plant mentioned in the literature.

The last model of a harmonic drive was made by one of the authors of this book – V. Tarabarin – in 2004 (Fig. 3.97). The complete set of details made by post-graduate O. Filippov served as a basis for this model [73]. Filippov’s thesis concerned the mechanism of a slave manipulator control (control handle) with harmonic drives. The model has small dimensions and it cannot be used for demonstration purposes in the

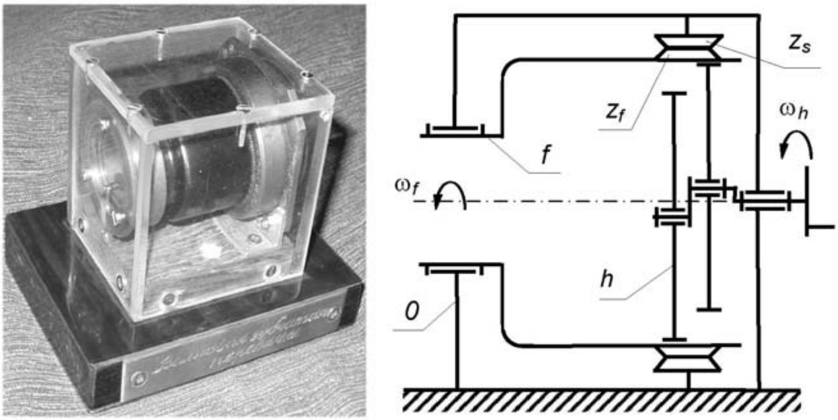


Fig. 3.97: The harmonic drive with a flexible-glass and a double-disk wave generator

lecture-hall. The wave generator of the mechanism is a disk – the diameter of the generator's disks of the generator is a little less than the internal diameter of the flexible gear. The disks are located on the input shaft on two eccentrics. The eccentricities of the discs have opposite directions. The value of the eccentricity of the disks in such a wave generator is small, therefore it has a small moment of inertia. The flexible gear is of glass. The face flange of the flexspline is small as there is an aperture with a rather big diameter in its nave. The module of the gearing m = 0.3 mm, teeth numbers: of the rigid gear z_s = 140, of the flexible gear z_f = 138, the wave generator shaft-to-flexible-gear shaft gear ratio at a stopped rigid gear u^s_{hf} = 69. This aperture does the work of observing the disk generator. There is a light-emitting diode inside the flexible gear used for illumination, it is switched on by pressing the button located on the pedestal of the model. To make the internal structure of the mechanism visible, the case of the model is made of transparent plastic.

A harmonic drive of external deformation is the invention of TMM department members V. Tarabarin and G. Timofeev [74, 75]. The mechanism of this drive differs from the usual harmonic drive as its wave generator is located outside its flexible gear. Rings of the wave generator are installed on crankshafts and displaced from the central axis of the mechanism on eccentricity value. As a whole, the generator represents a twin mechanism of parallel cranks with lower pairs. The internal surfaces of wave generator rings form an elliptic surface. This surface deforms the flexible gear and meshes its internal teeth with external teeth of the rigid gear. The gearing in the external deformation drive occurs in the zone of minor axis of the deformed flexible gear. When the cranks of the generator rotate, its ring makes a circular translational motion. The radius of these circles is equal to the eccentricity of the cranks. When crankshafts rotate, the deformed curve rotates synchronously with them. The gear ring of the flexible gear rolls over a rigid gear ring. The rigid gear in this drive has external teeth and in the design being examined, is fixed on the output shaft. The external deformation drive is not a coaxial one, so the axis of the engine does not coincide with the axis of the output shaft. In drives of radar antennas, cables and waveguides pass through the shaft of the drives, so a central aperture of a large diameter is necessary in drives. It is easier to provide it in a non-coaxial drive. The number of input shafts of the generator is usually not less than three; therefore drives of external deformation are suitable for the creation of multiengine drives. Large friction loss in a generator ring-flexible-gear pair and a large number of redundant constraints in the mechanism of the wave generator, are the main disadvantages of these gearings.

The model of a harmonic drive of external deformation was made as an enterprise with which the TMM department kept the experimental-design work to the application of such drives of radar antennas in 1975. A photo of this model and its kinematics scheme are shown in Fig. 3.98. The model has three crankshafts and only one shaft is power-driven. The flexible gear is made of a glass and fixed in the case. The rigid gear is made integral with the output shaft of the drive. Teeth numbers: of the rigid gear z_s = 300, of the flexible gear z_f = 302, wave generator shaft-to-flexible-gear shaft gear ratio at a stopped rigid gear u^f_{hs} = 150.

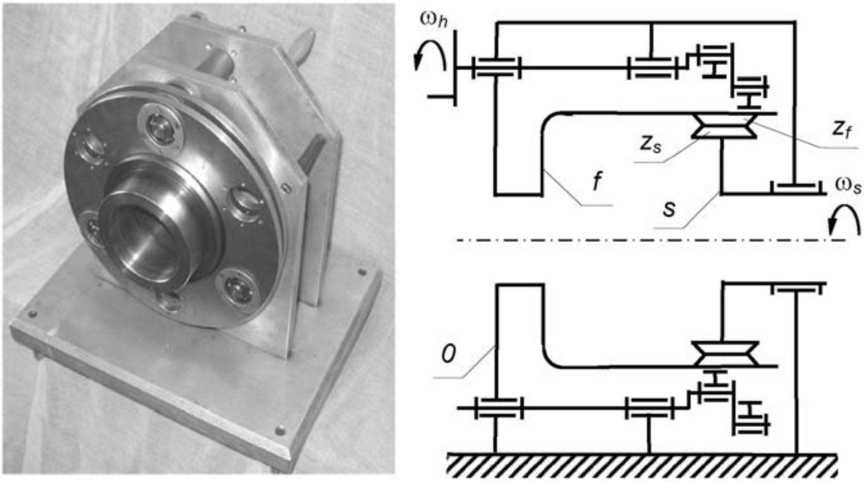


Fig. 3.98: The harmonic drive of external deformation with a flexible-glass and a double-ring wave generator

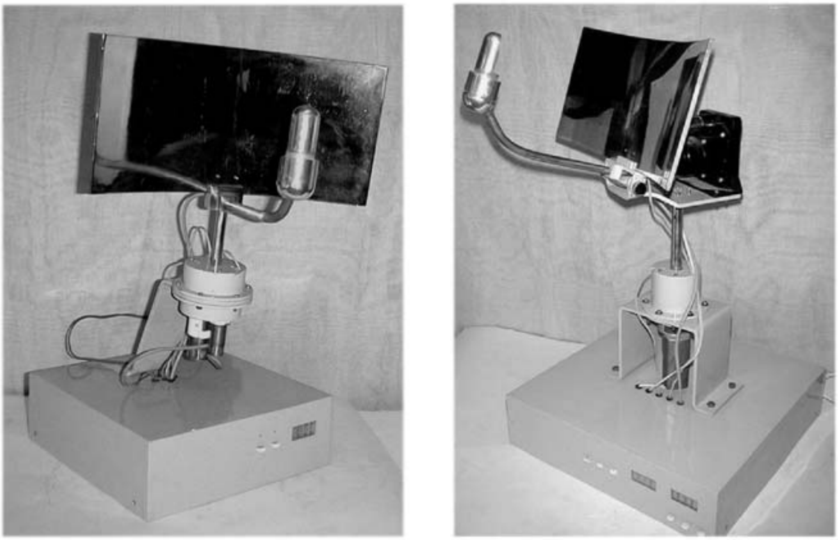


Fig. 3.99: Breadboards of aerials with tooth harmonic drives

From 1965 to 1990 the TMM department kept the experimental design and research works on harmonic drives and planetary trains. As the result of these works, the department gathered a significant collection of full-scale models of harmonic drives. Some of them are presented in the collection of mechanisms of the department. Breadboard models of antenna devices with harmonic drives are shown in Fig. 3.99. In the left photo there is only one drive with a harmonic drive – it is a drive of antenna

rotation relative to its vertical axis. The drive is electromechanical, the harmonic drive has a flexible-ring and a harmonic drive clutch. The gear ratio of the harmonic drive u = 80, the module m = 0.4 mm. In a breadboard model in the right photo, two drives with harmonic drives are used. It is an antenna turn drive and an antenna reflector swinging drive. In the turn drive, a harmonic drive with a flexible-glass with the gear ratio u = 80 and module m = 0.3 mm is used. In the swinging drive, a combined mechanism that includes a harmonic drive and a screw-nut transmission is used. The gear ratio of the harmonic drive u = 80, module m = 0.3 mm.
In 1990, a group of members of the TMM department, in collaboration with radio design office members, created a combined planetary-harmonic drive mechanism [76]. In this mechanism, a crank-planetary mechanism and a harmonic drive of external deformation are connected sequentially to each other. External teeth are cut on the rings of the wave generator. The rings of the generator are at the same time the satellites of the planetary mechanism. The gear ratios of the planetary train and the harmonic drive are equal to each other. The fixed gear of the planetary train and the rigid gear of the harmonic drive are connected to the output shaft. The module of the planetary train is more than the module of the harmonic drive. In the drives of the mechanism, it is necessary to coordinate the modules, and also to tolerate equality of

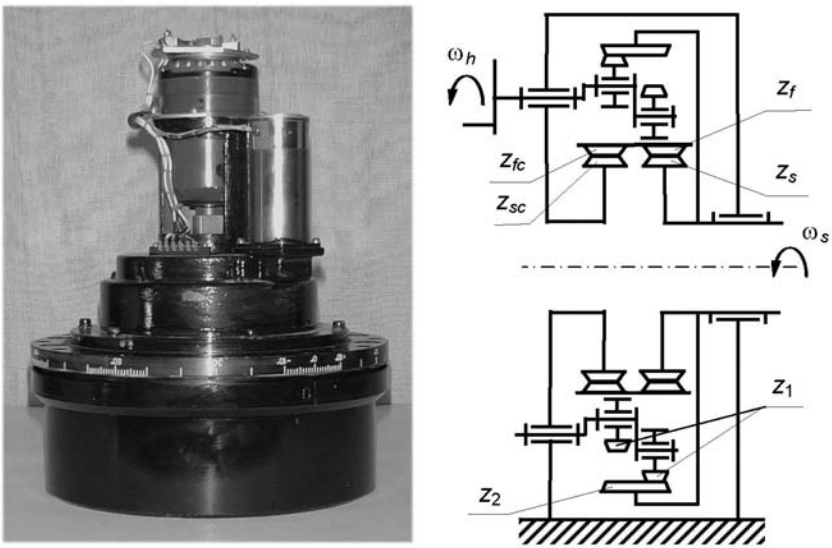


Fig. 3.100: A combined planetary – harmonic dear drive

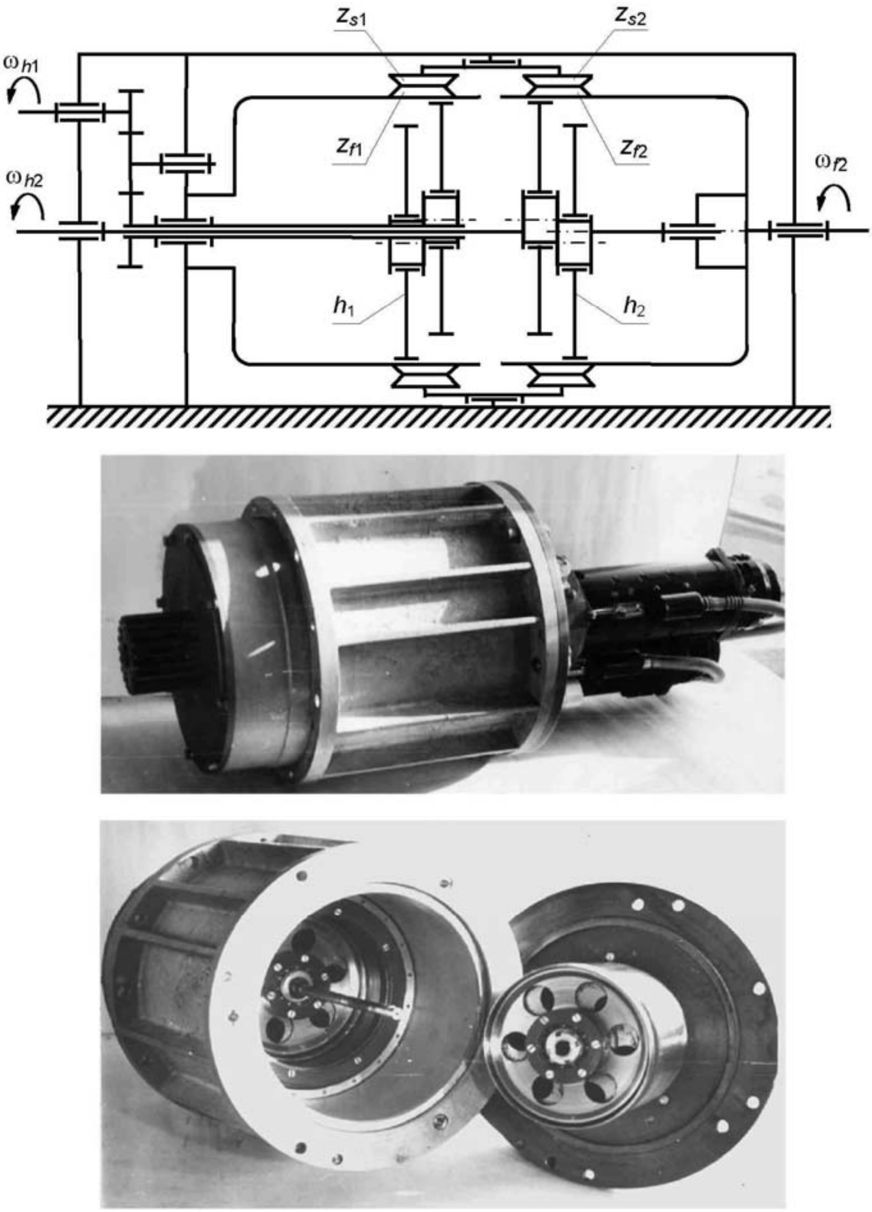


Fig. 3.101: A differential drive with tooth harmonic drives of internal deformation with disk wave generators

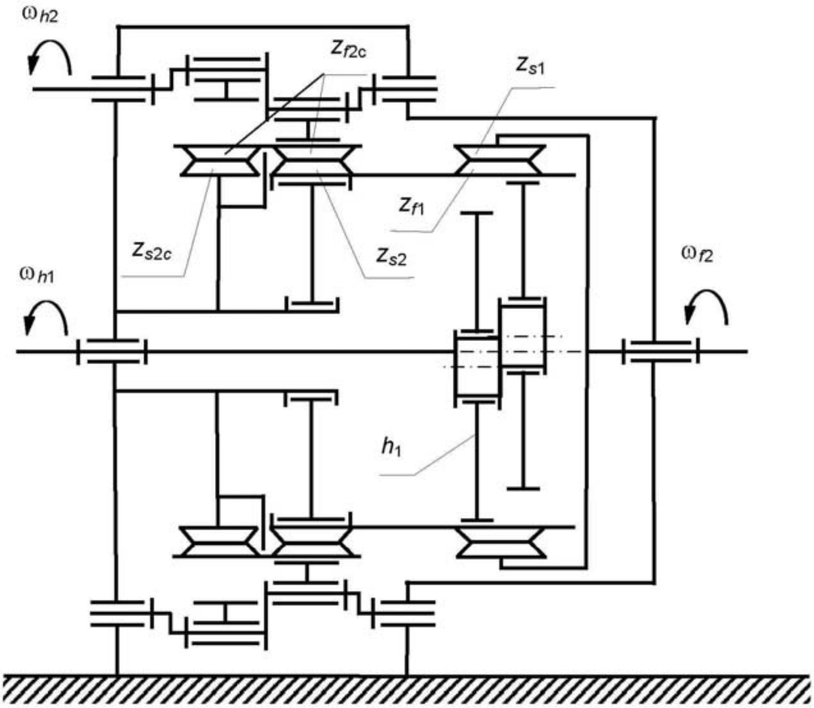


Fig. 3.102: A differential summing drive with a harmonic drive of internal deformation with the wave disk generator and with a harmonic drive of external deformation

interaxal distances (eccentricities) and gear ratios. In comparison with a planetary or a harmonic drive, the combined mechanism has higher accuracy, rigidity and loading capacity. Due to the use of a greater number of mashing zones, the weight and dimensions of the mechanism can be reduced essentially. A pre-production model of the combined mechanism was made at one of the enterprises of Rostov-upon-Don (Fig. 3.100). It was transferred to the BMSTU's TMM department for testing. Nowadays the mechanism is kept in the BMSTU collection of mechanisms.

In 1973, the research section of the TMM department worked in collaboration with the Design Office "Instrument-Making" on designing and research of harmonic drives for two-channel control systems. The aim of these drives was the summation of movements of two controlling engines on its output shaft. A harmonic drive differential was offered by Ju. Sinkevich [77] in 1967. Postgraduate V. Tarabarin designed two mechanisms: the first one was designed according to the scheme offered by Sinkevich the second one – according to the one offered by Tarabarin. In the first mechanism, two harmonic drives of internal deformation with disk-wave generators are used. In the second mechanism a harmonic drive of internal deformation with disk generator and a harmonic drive of external deformation with a wave ring generator are used. Preproduction models of these mechanisms were manufactured. The research of the drives was the subject of Tarabarin's thesis [78] in which definition tests on torsion rigidity, backlash, free frequencies and efficiency of the drives were conducted. Nowadays, these mechanisms are kept in the TMM department's collection of models.

A differential mechanism with harmonic drives of internal deformation and its kinematics scheme are shown in Fig. 3.101. The number of teeth of the wheels of the mechanism: circular spline $z_s = 242$, flexspline $z_f = 240$. Gear ratios of harmonic drives $u^s_{hf} = 120$.

A scheme and photos of the second drive are shown in Fig. 3.102. In the first channel of the drive, a drive of internal deformation with a flexible-gear-pipe and a disk wave generator are used. In the second channel, a harmonic drive of external deformation with a flexible-ring and a harmonic drive clutch are used. On the flexible-gear-pipe two gear rings are cut: a gear ring of harmonic drive of internal deformation z_{f1} (scheme on the right) and a gear ring of external deformation drive z_{s2} (scheme on the left). The left ring z_{s2} bases on a disk installed in the case supported in bearings. This ring meshes with the flexible-gear-ring with internal teeth z_{f2}. The right gearing formed by gears z_2 and z_{s2c} is a harmonic drive clutch in which the rigid gear z_{s2c} is fixed in the case. The number of teeth of the wheels of the mechanism: $z_{f1} = z_{s2} = 240$, $z_{s1} = z_{f2} = z_{s2c} = 242$. Gear ratios of harmonic drives (at braking of shaft one of the engines) $u^s_{hf} = 120$. Both mechanisms are intended for use in one device. The first mechanism has bigger weight and dimensions, has greater accuracy, better dynamic parameters and higher efficiency.

3.4.3. Spatial gearings

3.4.3.1. Non-involute gearings

Gearing with dotty contact (for example, cylindrical tooth wheels with conic wheels) named "non-involute gearing" by Ya. Davidov [79], were projected and designed by TMM postgraduate. Conic wheel teeth in such gearings are machined with a ram instead of a cutter head. At the same time in fact, a "conic wheel" is a cylindrical tooth

wheel machined with variable displacement (tool displacement changes in linear law). Tooth profile surfaces of such a wheel would be cylindrical involute formed from a base cylinder instead of a base cone. The second wheel of this gearing would be a usual cylindrical tooth wheel. Divers of mechanisms of this kind are presented in the model collection. First gearings (Fig. 3.103) with non-involutes engagement were designed by postgraduate Ya. Davidov [79] in the TMM department of BMSTU in the late 40s.

He write in his memories about creating this model: “The device wasn’t intended for creating precision wheels. I had modest funds, that is why only confirmation of the design theory seemed possible. Some straight tooth wheels were cut including hypoid ones.

But there were neither spiral ways for cutting oblique tooth wheels nor an oblique tooth ram in the workrooms, that is why oblique tooth wheels weren’t cut. Conical wheel nor was a cut to get nonorthogonal gearing. Another device was necessary for it. The Ph.D. thesis “Spatial gearing of tooth wheels with non-involute profiles cut by an involute cutter” was defended by Davidov in BMSTU in the spring of 1948. Later on, these gearings were worked by postgraduate students K. Bogolubsky, E. Soldatkin, I. Sekey, A. Efimenko, N. Prohorova and others.

Bogolubsky had been taught at a postgraduate course simultaneously with Davidov. He studied non-involute gearings too. In Fig. 3.104, photos of four models projected from the results of Bogolubsky’s thesis [80] are presented.

Fig. 3.103: The model of orthogonal conical non-involutes gearing created by Ya. Davidov

Here we can see that Bogolubsky overcame the difficulties that Davidov wrote about. He created a model of nonorthogonal conical gearing and cut oblique tooth wheels. Model photos are presented in his thesis and we can presume that they were created in 1948–1950. Model L-13 is a spatial conic gearing with an intermediate wheel. The input wheel of the model is a cylindrical straight toothed. Two other wheels are conic straight tooth

wheels machined by an involute ram at variable displacement. Model L-14 consists of three tooth wheels too: one cylindrical and two conic. Conical wheels are placed on the output shaft and the cylindrical wheel is placed on the input shaft. The cylindrical wheel is placed on a dowel and can move along the shaft axis direction.
While moving along the shaft, the wheel can be entered into the gearings either via the first or second conic wheel. Teeth numbers of conic wheels are different, that is why the reduction ratio of the mechanism changes after switching. Two other models shown at Fig. 3.104 are gearings with oblique tooth wheels. In the right model, a cylindrical straight tooth involute tooth wheel comes into action with the conical oblique tooth wheel. In the left model, a straight tooth conical wheel comes into action with a cylindrical oblique tooth involutes wheel. The last model didn't survive, its photo is taken from Bogolubsky's thesis.

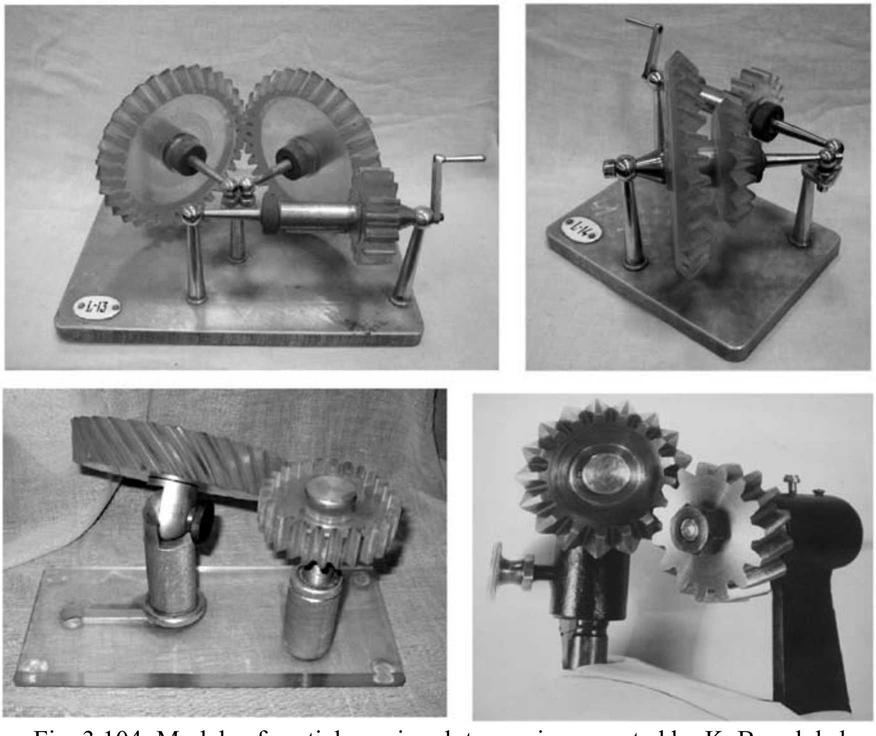


Fig. 3.104: Models of spatial non-involute gearings created by K. Bogolubsky

The thesis written by postgraduate student A. Efimenko [81] was about non-involute gearings too. It considered hyperboloid gearings of internal toothing. Based on the thesis results, model shown in Fig. 3.105 was created. A working pair and experimental assembly are also shown in the picture. The model was created in the TMM department of BMSTU in 1968–1969 (thesis was defended in 1969). Models shown in Fig. 3.106 are non-involute gearings. Neither authors nor date of production is known. The right photo is a model of conic gearing with a variable reduction ratio.

The input tooth wheel of this mechanism is straight tooth cylindrical. The ring gear of the output conic wheel is cut on the cone surface along a closed curve. One part of the ring gear in placed near the top of the cone and the other one is near the base. The reduction ratio is variable. Through one revolution, it changes through a periodical principle. When the teeth interlock near the top of the cone, reduction ratio of mechanism is minimal. When zone of meshing traverse to the base of cone, the reduction ratio increases. Gearing with variable reduction ratio was developed by postgraduate student K. Tarhanov [82].

The left photo shows the mechanism with an output link screw motion. It uninterruptedly turns and periodically traverses along the axis. Through one revolution of the output link, it makes some full oscillations in the axis direction. One can suppose that these models were developed and created by postgraduate student of Professor Ya. Davidov. These postgraduates produced their works in other Institutions and defended them in the Science Council of BMSTU. After defending, the models were presented to the TMM department and so enriched its collection of mechanisms.

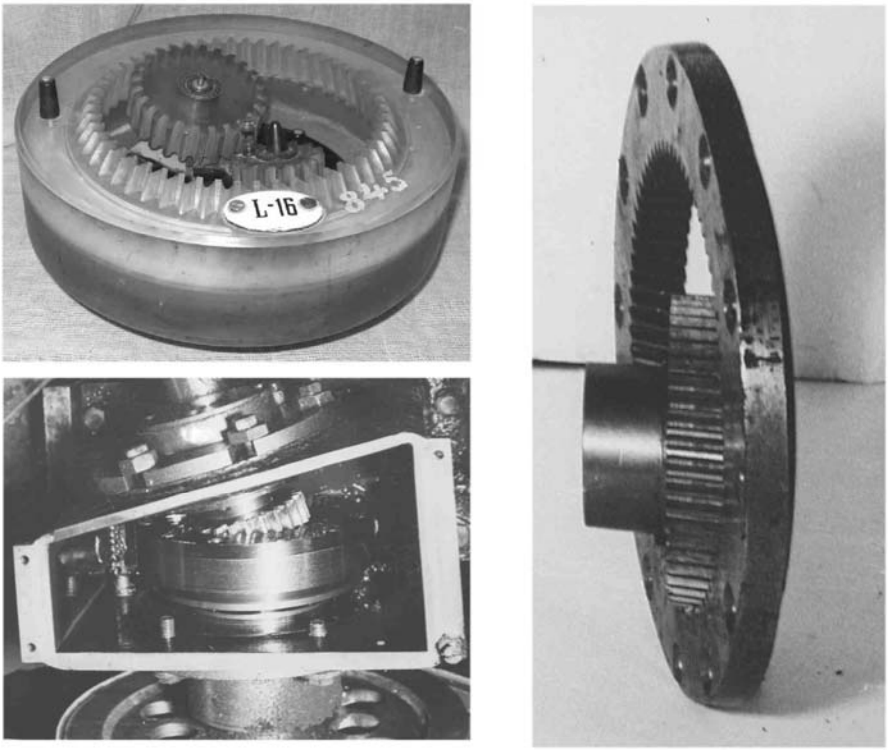


Fig. 3.105: Model of “non-involute” conical internal gearing and experimental assembly for testing such gearing

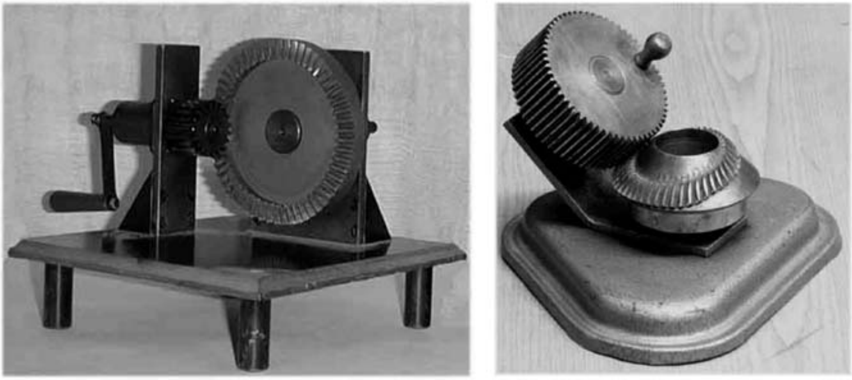


Fig. 3.106: Models of “non-involute” conical gearings with screw motion of output link (left side) and with variable reduction ratio (right side)

3.4.3.2. Gearings with variable angles between axes

“Non-involute gearing” is also used in gearings with variable angles between wheel axes. These gearings were designed and examined in the Candidate [83] and Doctoral theses of E. Soldatkin. There, one or both wheels are machined with the use of dual parameters. One of the parameters receives the formation of the teeth’s surfaces according to the relative motion equivalent to its turning around axes. The second enveloping parameter forms surfaces receiving meshing with an uninterrupted changing angle between the axes of the wheels. Created meshing has a dotty contact of wheels. Changing the axis angle can be made simultaneously with transmission of the angular motion. Such gearings can be successfully used in manipulators for motion transmission by movable joints of links. There are two models of such mechanisms in the collection.

a. parallel wheel axes b. wheel axes intersect at arbitrary angle c. wheel axes are coincide

Fig. 3.107: Model of toothed muff with a variable axle angle (at variable axle positions)

The model shown in Fig. 3.107 has a reduction ratio equal to 1 and allows changing the angle between wheel axes from 0 to 180 degrees. The number of teeth of the wheels are equal to $z_1 = z_2 = 22$. The teeth of the wheels are fixed on the cylindrical outside surface with partial transition to the end surface plane. Such constructions of

wheels get their meshing in the whole diapason of a changing axis angle. This mechanism can be used as a universal joint for connecting revolving shafts with a variable angle between axes. Changing the axis angle change is achieved by turning the bearing body of one wheel relative to the other. So, bearing bodies of tooth wheels are connected by a revolute pair. The axis position of this pair should be matched with the enveloping parameters used while making the wheels.

a. parallel wheel axes b. wheel axes intersect at an arbitrary angle c. wheel axes are

Fig. 3.108: Model of a toothed muff with a variable axle angle (at variable axle position)

The second model (Fig. 3.108) is formed by meshing the cylindrical involutes wheel with a wheel machined by the method of dual parameter enveloping. The reduction ratio of the gearing is $u_{12} = 0.77$, the number of teeth of which are $z_1 = 17$, $z_2 = 22$. The changing axis angle can be changed from 0° to 90° in this mechanism. Only the teeth of the second wheel are machined using dual parameter enveloping in this model and only these wheel teeth transit to the end surface plane. The second wheel with a smaller teeth number is usually a cylindrical involute one.

3.4.3.3. Worm, screw, bevel and spiroid gearings

Worm and screw gearings are represented in the collection mainly by the models of Reuleaux and Redtenbacher. But some models were created in a later period – in the 1970s. Figure 3.109 shows two models of spatial gearings: a model of worm gear with a cylindrical worm (left) and a model of a double-reduction gear. The first stage of which is the worm and the second one is the screw (right one). These models were created under the direction of Professor A. Golovin by students of the BMSTU branch in Kuntsevo. The models are for the demonstration of spatial gearings for students.

One more hyperboloid gearing model (Fig. 3.110a) was presented to the department by one of the postgraduates. The creator and the place of creation of this model are not known. The gearing of this model is placed between worm and screw mechanisms. A large canting angle of the screw line is usual for worm gearings. From the other point, the diameter of the worm is much smaller then the wheel diameter in worm gearings. Vice versa, the input wheel diameter is larger than output wheels in this model.

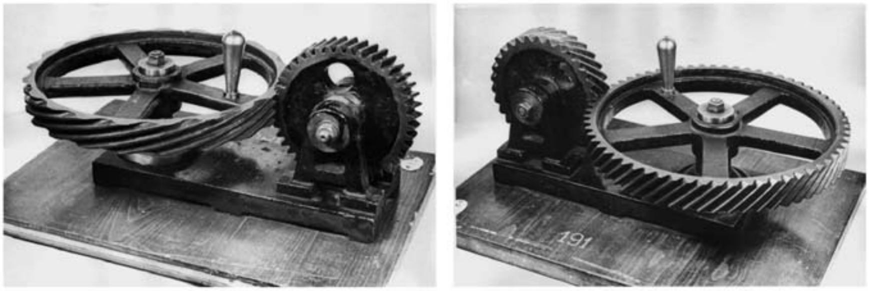
Fig. 3.109: Models of spatial hyperbolical gearings made by BMSTU students

Two models of screw tooth gearings are similar in constructive registration (Fig. 3.110b). In these models the diameter of the entrance cogwheel too is more than diameter of the output wheel. Models among themselves differ numbers tooth and a angle of an inclination of a line of a tooth. In the model located in Fig. 3.110b, a angle of an inclination of a line of a tooth on an entrance wheel about 70°. The number of tooth wheels are z_1= 20 and z_2 = 40, the module of gearing m_n = 4 mm. In the model in Fig. 3.96c the angle of inclination of lines of a tooth of cogwheels is equal to 45°. The numbers of tooth wheels z_1 = 36 and z_2 = 62, the module of gearing m_n = 4 mm. These models have large dimensions and weights, therefore their use for demonstration at lectures is impossible. It is possible to assume that these models were made at BMSTU approximately in 30 years of the last century.

Bevel tooth gearings are also in the collection, both models of tooth gearings, and complete sets of cogwheels. In Fig. 3.111a three complete sets of cogwheels are shown: two pairs of conic wheels and one pair of flat wheels. All these wheels have the line of a tooth executed on an intricate curve. These wheels can be approximately determined as bevel wheels with the herringbone line of a tooth. The time and a place of manufacture of these cogwheels are not known.

Bevel tooth drivers with a circular tooth are widely applied in transport machines as the main gears. These gears allow connecting the shaft crossed under an angle of 90°. They provide a large reduction ratio, high smoothness of the job, and low noise level. For manufacture bevel cogwheels with a circular tooth, a special tool and special machine tools are necessary. These cogwheels often cut gear-cutting machines on Glisson's firm. A process cut tooth on these machine demands the performance of a difficult engineering setup of gear-cutting machines. The model of bevel tooth drivers with a circular tooth (Fig. 3.111b) was made in the workshops of the TMM department approximately in the 50s and 60s. The number of tooth wheels of the model are z_1 = 14 and z_2 = 26, the external module of gearing m_e = 12 mm.

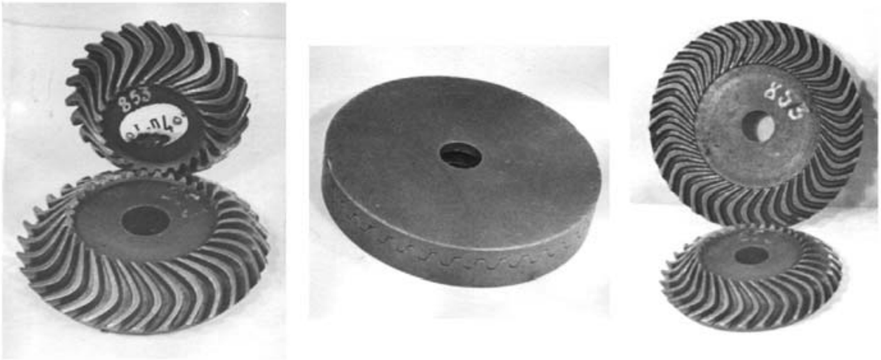
a. Model of spatial hyperboloid gearing



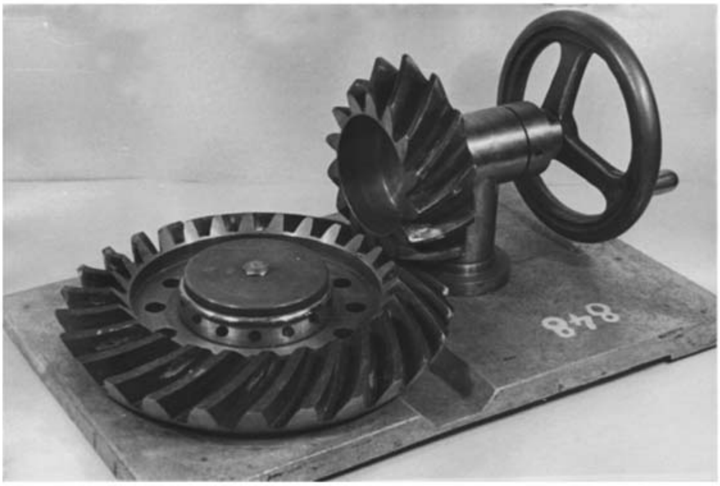
b. Models of screw spatial gearing with crossed-axis

Fig. 3.110: Models of spatial hyperboloid gearing

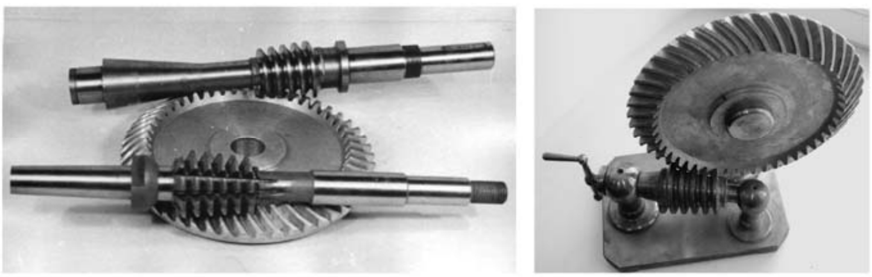
Spiroid gearings (Fig. 3.111c) are placed between conical hypoid and worm gearings. In comparison with conic gearings, the spiroid ones have a larger reduction ratio, because the pinion may have only one tooth. They are easier in the regulation of the contact patch, less sensitive to mounting errors, and less noisy. They don't need the console installation of a pinion. In comparison with worm gearings, the spiroid gearings have a larger contact ratio. For example, if the wheel teeth number is $z = 50$, then the compact ratio approaches 6. Less sliding of profiles allows to use of cast iron for wheels instead of costly bronze. Spiroid gearing research in the TMM department of BMSTU was carried out by V. Kuzlyakina. Professor V. Gavrilenko was the science head of her work. Kuzlyakina defended her thesis in 1972 [84]. A model of spiroid gearing was created in the department earlier. Model construction allows us to relate its creation to the 1960s.



a. Bevel and face cogwheel with a herringbone line of a tooth



b. Model of a bevel drive with a circular line of a tooth



c. Model of a bevel spiroid gearing, a worm hob for its cutting and the complete set of spiroid cog-wheels

Fig. 3.111: Models of bevel gearing

3.5. Models of explosion engines

3.5.1. Wankel engine

Three models of a collection are devoted to explosion engines – to Wankel's and Stirling's engines. These engines still cause heightened interest. Now it is possible to obtain plenty of material on the history of the occurrence of these engines, about features of their designs and working processes, and about perspectives of their application in various areas. Materials from [85–89] are used in the description below.
First, we shall examine Wankel's engine (Fig. 3.112). The idea of the creation of a similar mechanism was expressed in the 16th century. But only in 1924 did Felix Wankel executed the first outline developmental works of a rotary engine design. The first patent for an explosion engine was received by Wankel in 1929. However, the design of such an engine was realized only in 1957. By this time, the design had become as it is known to us today.
At this time Wankel worked as an engineer in the auto-motorcycle company NSU. He managed to convince the company management of the perspectivity of his engine. Per 1967 at the Frankfurt motor show, the first car with NSU Ro-80 a rotary engine was presented. Because of this it received the title "Car of year". The patent for the manufacturing of rotary motors was bought by many automobile and engine-building firms. Unfortunately, the majority purchased only a license. Successes in the creation of cars with rotary engines came to Mercedes which produced the conceptual coupe C-111, and Chevrolet which established such engines on pre-production models of the Corvette. The NSU Ro-80 car produced for, was almost ten years, but didn't receive popularity and became the initiator of the company's crash and so was absorpted by Audi. The greatest successes in the introduction of rotary engines was achieved by Mazda and AvtoVAZ. Only these two firms produced cars with rotary engines.

Fig. 3.112: Wankel's engine

The basis of the rotary-piston Wankel engine was the planetary mechanism with cycloidal gearing. In this mechanism the triangular rotor (three-teeth wheel with external teeth) was established as the eccentric (planet carrier) on the output shaft. The rotor can rotate around the eccentric. The stator (a cogwheel with two internal teeth) is motionless. For transferring the rotary movement of a rotor to the output shaft of the engine, involute the gearing with internal gearing is used. The transfer ratio of this transfer u_{21} = 2/3, that is the same as the relation of teeth numbers of rotor and stator. The form of the profile of the internal cavity of the case is epitrochoid. There are three cavities formed between the rotor and the case the volume of which changes at the rotation of the eccentric. In Wankel's engine, these cavities are combustion chambers in which a cycle similar to the cycle of piston engines is realized. At the combustion of fuel, forces of gas pressure influence the surface of the rotor and force it to turn the shaft of the eccentric.
Let us consistently consider the processes occurring in the chambers of the rotary piston engine. In Fig. 3.113, four positions of a rotor (through 90°) are represented. We shall designate chambers by the letters a, b and c. In position 1 in chamber a the process of outcome of gasses ends and the working mix suction process begins, in chamber b there is a process of working mix compression, and in chamber c there is a process of expansion. In the second position: in chamber a there is a suction process, in chamber b the compression process proceeds, in chamber c the exhaust gases outcome begins. In the third position: in chamber a the suction process comes to an end, in chamber b the working mix is firing and the process of expansion begins, in chamber c outcome proceeds. In fourth and last position: in chamber a suction process comes to an end, in chamber b expansion proceeds, and in the chamber c there is an outcome process.

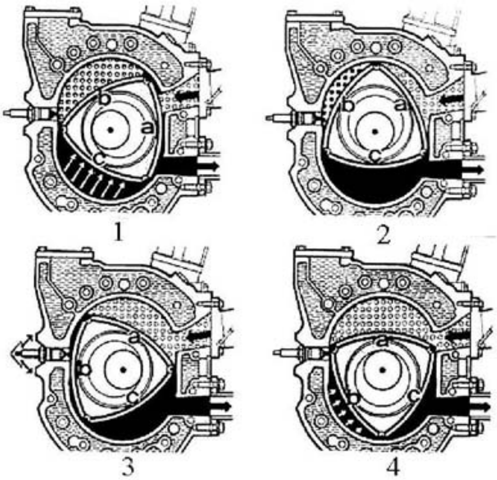


Fig. 3.113: Phases of working process in a rotary Wankel engine

Wankel's engine has no valve gear system, there are no pistons and connecting-rods. The engine has a small number of details and small weight. At identical capacity, these engines have sizes two to three times less than similar piston engines. In Wankel's engine it is necessary to counterbalance only the eccentrically established rotor; it is enough to establish counterbalances on the output shaft for this purpose. During the creation of this engine's design, there were difficulties with encapsulation of sealing disks of the rotor, with channels of gaseous exchange systems, carburation, etc. Wankel's engine has the form of a combustion chamber negative from a position of thermal losses. It is the reason for low display efficiency and explains the large fuel rate. Absence of translational moving weights, a smaller number of friction surfaces and bearings improves the mechanical efficiency of the engine and partially compensates for greater thermal losses. The contents of harmful substances in the exhaust gases of the rotary-piston engine has both advantages and disadvantages. Advantages are the presence of lower temperatures of combustion in the rotary-piston engine causes a smaller quantity of NO_x. Conditions for the formation of CO and CH_x are similar to the conditions in piston engines. However, the greater surface of the combustion chamber and greasing of a rotor working surface by addition of oil in the fuel (as in two-stroke engines) lead to the raised quantity of CO and CH_x.

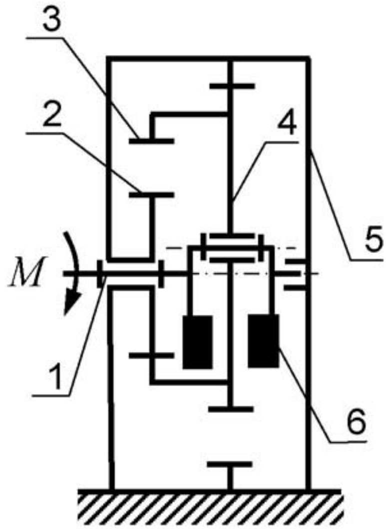


Fig. 3.114: Structural scheme of the KKM type Wankel engine model

In the DKM's motor, the rotor rotates around a motionless eccentric. Besides, in this type of engine, the internal part of the case surface also rotates. The main lack of the DKM engine consists in the complexity of the replacement of spark plugs. For this purpose, it should be completely disassembled. In 1954 Wankel patented the principle of duality of rotation according to which it is possible to invert the kinematics of the motor for a four-stroke rotary explosion engine. In the case of the KKM motor (Fig.

3.114) is motionless and the rotor sets in motion the shaft of the eccentric. The engine of KKM type has a number of advantages compared to the DKM engine: it is easier in manufacture, repair and service, is more compact, and the system of inlet-outlet works better in it. The KKM engine's birthday is July 7, 1958 which was when the first experimental sample was tested. The model of Wankel's engine was made in the workshop of the TMM department at BMSTU some 60 years ago by the educational masters of the faculty.

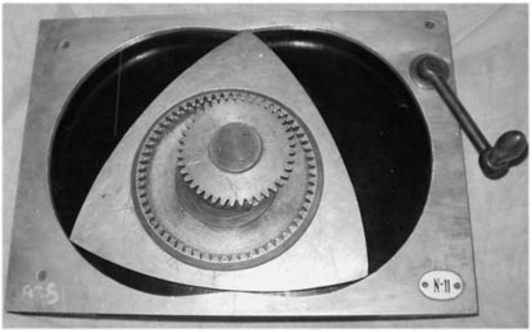


Fig. 3.115: Rotary Wankel engine model

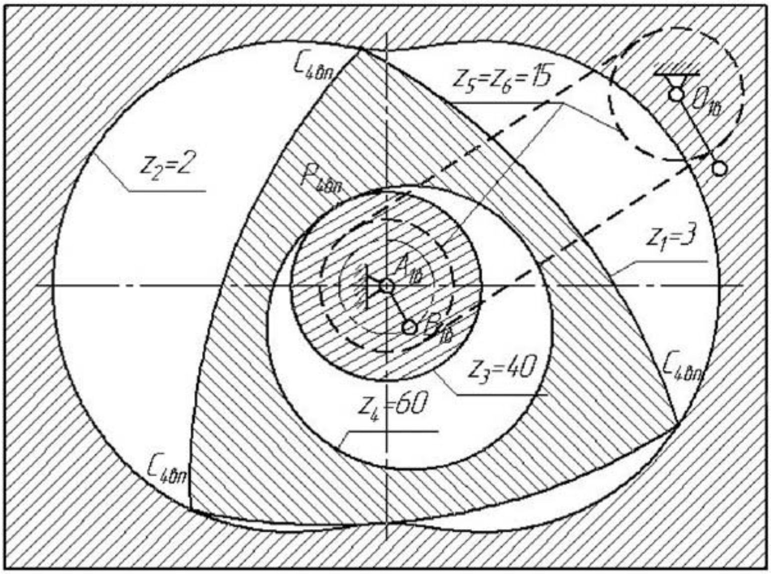


Fig. 3.116. Kinematics scheme of a rotary Wankel engine model

A photo of the model is shown in Fig. 3.115, and its kinematics scheme is shown in Fig. 3.116: Movement is transferred from the eccentric of the planetary mechanism to

a rotor in the model. In the model, eccentric AB is connected with the drive handle by a chain gear with the numbers of sprocket teeth $z_5 = z_6 = 15$. Rotation from the handle is transferred to the eccentric which sets in motion the rotor of the engine. The internal gear wreath with $z_4 = 60$ teeth revolves around the motionless wheel with external teeth $z_3 = 40$. As a result, the plain-parallel movement is set to a rotor. The scheme of the model corresponds to the KKM type engine with a motionless case. In a real Wankel engine the rotor makes a plain-parallel movement under the pressure of combustion of the fuel products. This movement is transferred by means of a three-linkage planetary mechanism to eccentric AB which is the output shaft of the engine.

3.5.2. Stirling's engine

Robert Stirling (1790–1878), a Scottish pastor and doctor of divinity, occupies a place of honour among founders of thermodynamics and heating engineering [88, 89]. In 1816, Stirling received the patent for "the machine which makes motive power by means of heated air". In 1845, at a foundry in Denmark Stirling's first engine with a power of 50 hp was tested, it worked for about three years. For several decades there was no mention of Stirling's engine. Only in 1890 were such machines of low power produced. The creation and development of explosion engines forced out Stirling's engine and they were almost forgotten. Revival of interest in these began approximately in 1938. Scientific research was carried out and industrial development began. Stirling's engine is a machine hat works on the closed thermodynamic cycle in which cyclic processes of compression and expansion occur at different levels of temperature, and the management of the stream of a working body is carried out by a change of its volume.

The principle of Stirling's thermal engine action is: compression of a certain quantity of gas at low temperatures and the expansion at high temperatures. However, heating in Stirling's engine occurs in an absolutely different way: heat to gas is brought from outside, through a partition of the cylinder, therefore it is often named "the external combustion engine". In Stirling's engine a periodic change of gas temperature is used. The displacing piston moves the gas to one of two cavities in the cylinder. In one of the cavities, the temperature is low, in other one, a high temperature is constantly supported. During the movement of displacing the piston upwards, gas moves from the hot cavity into the cold one through a heater, regenerator and radiator. At downwards movement, the gas comes back the same way into the hot cavity. In the first case gas is cooled by heat transferring to a regenerator and refrigerator. In the second, it consistently heats up all over again in a regenerator, and then in a heater. The regenerator is intended for the reduction of heat losses, an increase of temperature difference of hot and cold gas, and the engine's general thermodynamic efficiency increases. It represents a vessel filled by porous material with high specific heat. During passage through the regenerator, hot gas heats up the material filling it and it is partially cooled. At the movement in the opposite direction, cold gas passes through a first regenerator, where it is partially reheated, reserved by a material in here. After the regenerator the warmed gas proceeds into a heater. A displacing piston is mechanically connected to the working piston which compresses gas into a cold cavity and transforms the kinetic energy of gas expansion into a hot cavity. As the compression of gas occurs in a cavity of lower temperature, then expansion, and useful

work is obtained. Four phases of the running cycle of Stirling's engine are shown in Fig. 3.117. Phase I – the working piston is in extreme bottom position, and displacing – in extreme top. All gas – in a cold cavity. Phase II – the displacing piston remains in the top position. The working piston compresses the gas at low temperature. Phase III – the working piston remains in the extreme top position, and the displacing one moves the gas from a cold cavity into a hot one. Phase IV – heated gas finishes the expansion, pistons (working and displacing) are in extreme bottom positions. While the working piston is in the bottom position, the displacing one moves the gas to a cold cavity. Then the cycle is repeated.

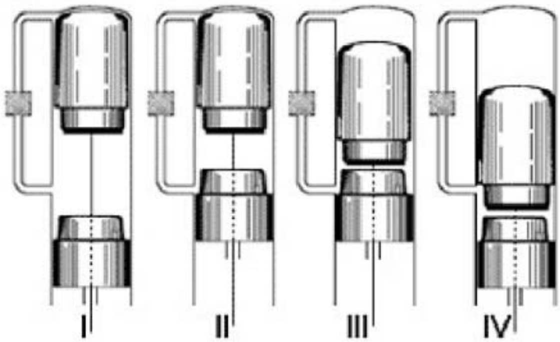


Fig. 3.117: The running cycle of Stirling's engine

For the absorption of heat it is possible to use any source of heat. As a working body in Stirling's engine, air, helium or hydrogen are used. The ideal thermodynamic cycle of Stirling's engine possesses the greatest possible thermodynamic efficiency and is 30–40%. The efficiency of the engine remains almost constant in a wide range of working conditions. Such values of efficiency can be received in the engine only at an effective regeneration.

Liquid fuel is applied in the majority of engines now. Usually, to heat a working body a continuous process of burning is used and that allows the burning of various kinds of fuel. At continuous burning, the level of hydrocarbon and carbon oxide emission decreases. In Stirling's engine, there are no valves and there are no periodic explosions in cylinders, so it is less noisy. The attitude of capacity to weight in Stirling's engine is comparable with similar parameters of a diesel engine with a turbo-supercharging. Specific capacity output is approximately the same per as in diesel engines. Fabrication cost of Stirling's engine is higher than the cost of explosion engines manufactory, however, working costs are less.

Now a lot of Stirling's engine designs are known. They differ in operating mode, way of pistons and cylinders connection. In a BMSTU's collection, there are two models of these engines. Both models concern mechanisms with teeth-lever rhombic drives. In Fig. 3.118a, block diagrams of teeth-lever mechanisms of Stirling's engine from work [90] are represented. Models of mechanisms are executed for one of these schemes are shown in Fig. 3.118b.

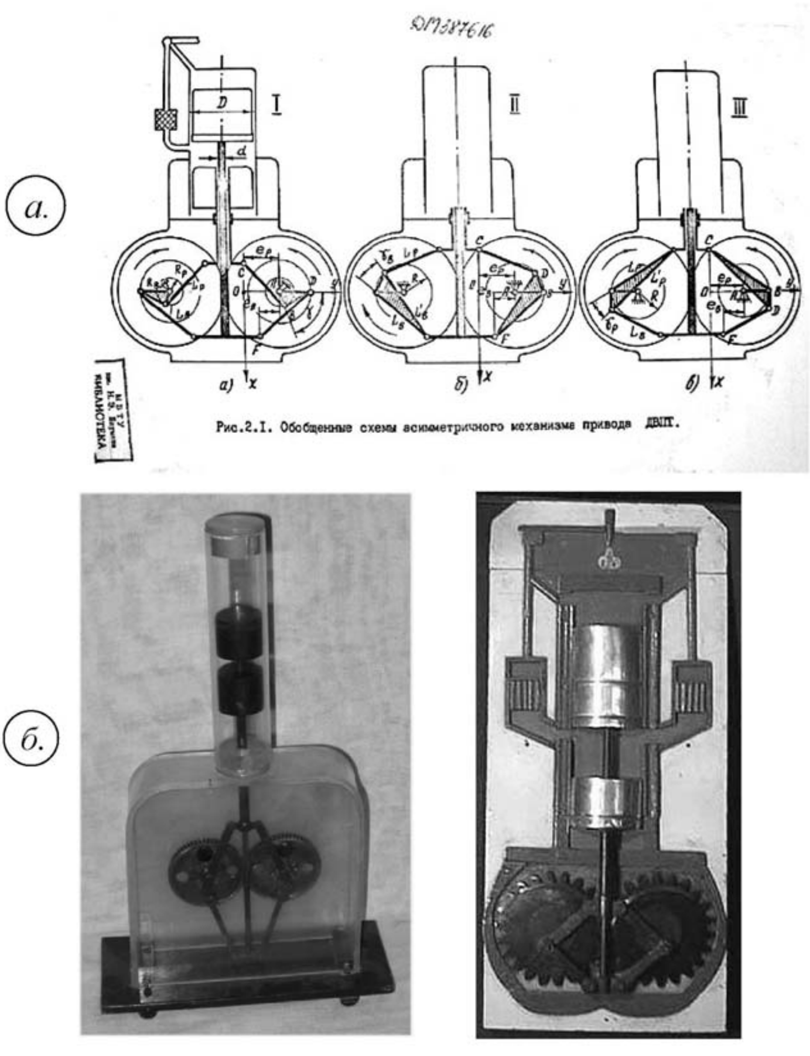


Fig. 3.118: Design schemes and Stirling's engine models

3.6. Models of typewriter printing mechanisms

Until the 1890s, shock action mechanical machines were basically used as typewriters. The first typewriters which allowed the production of words faster than writing by hand, appeared approximately in 1850 [91]. Since then the mechanisms of these machines continuously improved. In 1897 Underwood (USA) produced a typewriter with Wagner's printing mechanism which allowed the typist to see the text which was being printed. The design of this mechanism had three moving elements and compared with the printing mechanisms which were used at that time, was more simple and

convenient. In 1907 Torpedo (Germany) began to produce machines with a six-bar linkage mechanism. The design of this machine's transition from a font of the bottom register to a font of the top register was carried out by lowering the printing mechanism's segment. Wagner's mechanism change of the register was made by rising of the carriage. It had a significant advantage. As the weight of the segment was essentially less than the weight of the carriage. Besides such a design allowed the machines to be fitted with carriages of various sizes.
Requirements of a typewriter's mechanism are quite different. The mechanism should provide the demanded force of impact necessary for printing several copies of document. Thus, the text should be readable, not greased and not blurred. Therefore, it is necessary that during the moment of contact with the paper there are no extraneous movements, including vibration. Sufficient force of impact can be reached by increasing the mechanism's reduced mass and speed of the printed letter. For this purpose, it is necessary that the mechanism's transitions is increased at the end of the movement. Recommendations for the choice of printing mechanism parameters with various schemes are the result of [91]. The substantiation of these recommendations is given in Nemkevich's dissertational work [92] which was executed by him at the TMM department in 1965. A classification of printing mechanisms schemes is offered in this work. According to this classification, mechanisms are divided into five groups: I – mechanisms formed by the connection of two link gears; II – mechanisms formed by the connection of link gear and four-bar linkage mechanism; III – mechanisms formed by the connection of link gear and two four-hinged mechanisms; IV – mechanisms formed by the connection of two four-hinged mechanisms; V – mechanisms formed by the connection of three four-hinged mechanisms.
Three models of such mechanisms, based on the materials of this work, were made by educational masters. A photo of one of these models is shown in Fig. 3.119. Next to the figure of the experimental plan used by the author for carrying out research into the kinematics and dynamics of printing mechanisms.

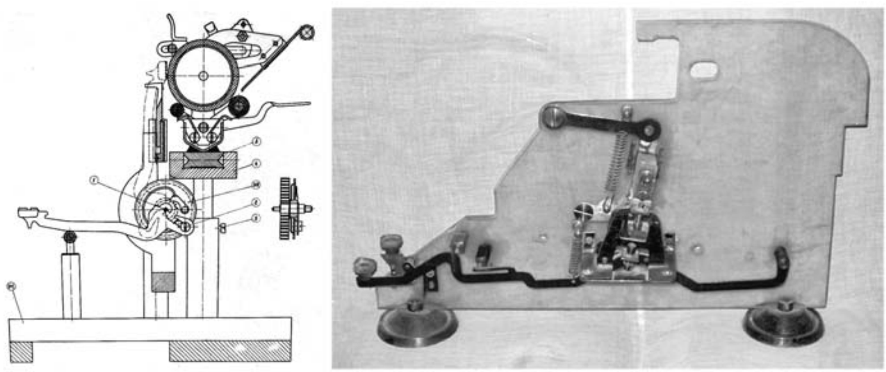


Fig. 3.119: The circuit of the mechanism of the printing machine (from [91]) and a photo of its model

The model of a six-bar linkage printing mechanism and its block diagram are shown in Fig. 3.120. This mechanism is related to the fourth group as it is formed by the consecutive connection of two four-hinged mechanisms. The next model of printing mechanism is the eight-bar linkage. It is formed by consecutive connection of three four-hinged mechanisms and concerns the fifth group. The scheme of this mechanism and its photo are given in Fig. 3.121.

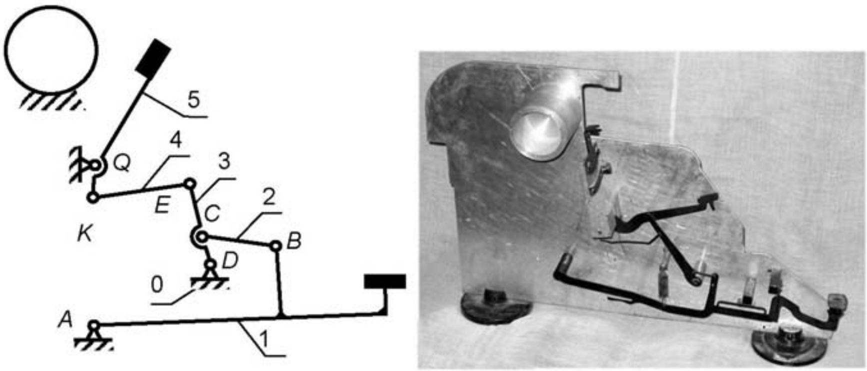


Fig. 3.120: The type diagram and photo of the six-links model of the printed mechanism

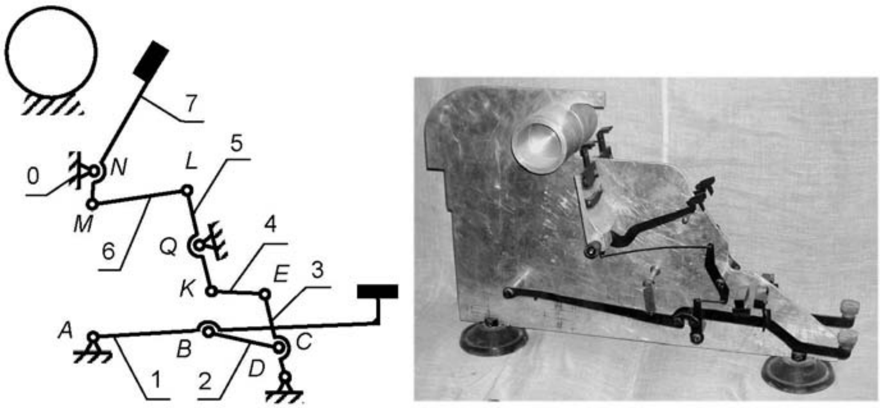


Fig. 3.121: The pyre diagram and photo of the eight-links model of the printed mechanism

3.7. Mechanisms of locomotive devices

3.7.1. Controllers

The TMM department of the BMSTU has been closely connected with steam-locomotive building (after locomotive engineering), steam machines and boiler fabrication departments. Professor L. Smirnov headed the TMM and steam machines departments simultaneously; Professor V. Gavrilenko headed locomotive building and TMM departments. Many teachers of the TMM department were graduates of the steam machines and locomotive building departments.

The departments conducted united scientific development the results of which have found reflection in models of mechanisms. So some models are based on the materials of the candidate dissertation of N. Pavlenko [93]. Pavlenko was a postgraduate student of L. Reshetov. He researched automatic driving gear mechanisms of electric contact devices of railroad rolling stock.

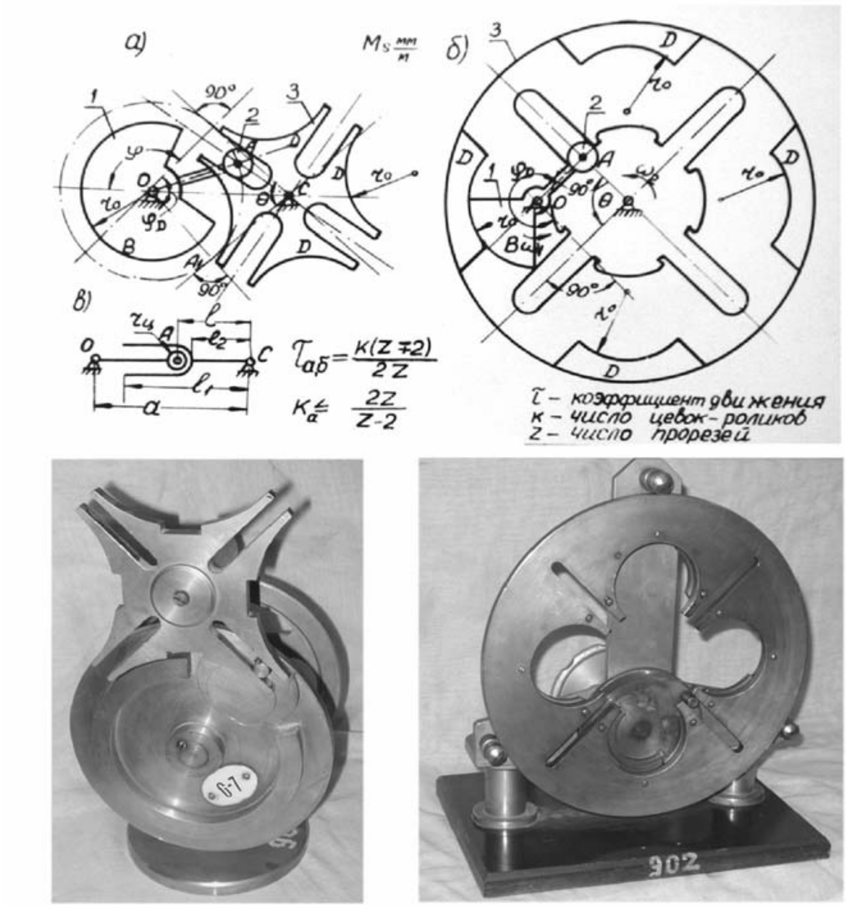


Fig. 3.122: Kinematics research design schemes of Geneva cross mechanisms and models of analyzed mechanisms [93]

Different multi-position control apparatus with Geneva, ratchet and cam mechanisms are observed in his work. Many of the developed mechanisms have been patented between 1959 and 1962 [94–99].
The kinematics of Geneva mechanisms with inside and outside gearing is examined in the dissertation and schemes and models are shown in Fig. 3.122. Automatic driving gears with Geneva mechanisms are used in switches of cascades of transformer EKG-60 (ЭКГ-60 in Russian) on electric locomotives N-60 (Н-60 in Russian), and also in control systems of electric locomotives VV-9200 (ВВ-9200 in Russian) of direct current and experimental electric locomotives N-80 (Н-80) with driving gear from the net of alternating current.
An electric driver with a cam-ratchet mechanism allows accurately fixing the output shaft in the required position. In this type of engine the cam shaft is rotating uninterruptedly on one side, and the orientation of the shaft's movement is changed by two pawls of a ratchet mechanism which are controlled by electromagnets. Selection of the cam's profile form allows the receiving of the required kinematics parameters of rotating links and the required principle of changing the reduction moment. A drive ensures high stability of the apparatus work both in position and in the transit regime. It totally prevents overtravel of positions or stopping between positions. Selection of the cam's profile allows an excepted hit at the moment of entering the cam's pawls in gearing with a ratchet wheel. The drive is compact, safe and simple to manufacture as it doesn't require high accuracy in details. The drive's scheme [93] and photos of its model are shown in Fig. 3.123 (the mechanism of pawl's switching is the large picture).
An electric motor drive with a reducer and locking ratchet mechanism (Fig. 3.124) is used in multi-position group controllers. In this drive, movements the parameters of the main controller shaft are set by the principle of the shaft's movement and transmission ratio of the reducer. The drive includes a conical differential and two locking ratchet mechanisms. This construction is used in the main controllers of French electric units of the ESM series. The drive ensures stable work of the apparatus both in positions and in transit. The main drive's disadvantages are its complexity and hits which appear during the fixing of ratchets by electromagnets. This ensures safe work in using low-powered and low-speed electric motors.
In Fig. 3.125, a multi-position Geneva mechanism is shown. As in the above-mentioned mechanisms this mechanism is used in automatic controllers of electric locomotives. The mechanism constructively combines a 15-n position Geneva mechanism and slider-crank mechanism for a lock. During the rotation of the output shaft the slider of the slider-crank mechanism leaving the mortice and trundle is rotating the cog-wheel by one angle step. After ending of the rotation, it comes into the mortice of the output drive wheel and fixes the frame. The mechanism is simple in construction and ensures exact position control of the output shaft and its safe fixing. Besides this, it has good kinematics and dynamic characteristics.
The mechanism in Fig. 3.126 is analogous to the previous one. It differs in the construction of the lock linkage in that it is a five-link jointed-level mechanism in which the roller of fixing lock is placed on the axis of moving rocker joint. This is the principle of the mechanism's work. When the trundle of the Geneva mechanism leaves the gearing with the mortice of gear wheel, the lock's roller comes into the cog-wheel's mortice and fixes it the frame. When the trundle comes into gear, the lock's

roller leaves the mortice and allows the rotation of the cog-wheel in one angle step. According to its qualitative factors, the mechanism is close to the mechanism of the crank-slider lock. The absence of translation pairs in the lock's mechanism is its advantage.

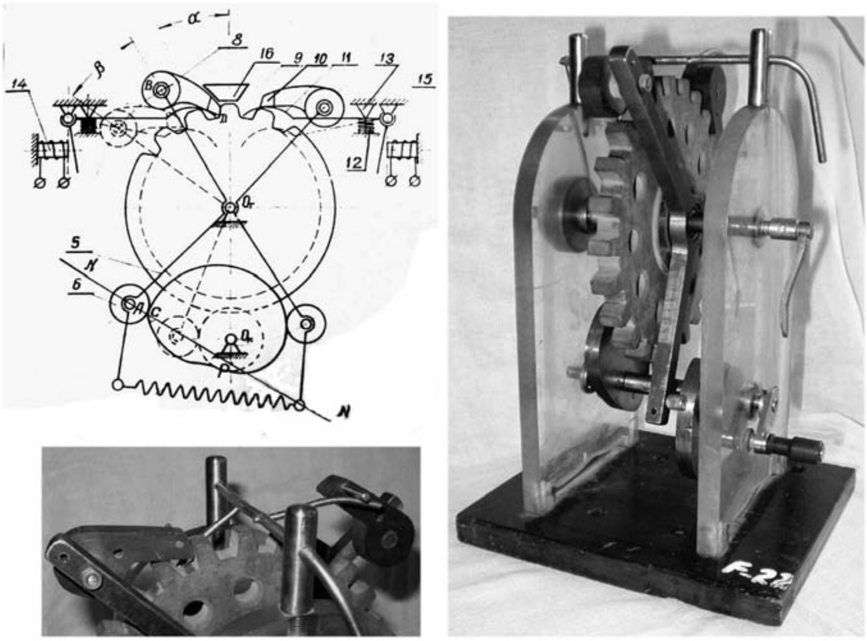


Fig. 3.123: A cam-ratchet mechanism with an electric drive

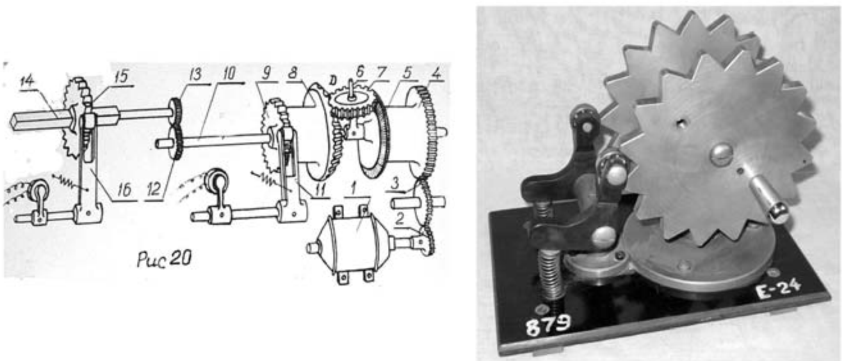


Fig. 3.124: An electric drive with a locking ratchet mechanism and model of the locking ratchet mechanism

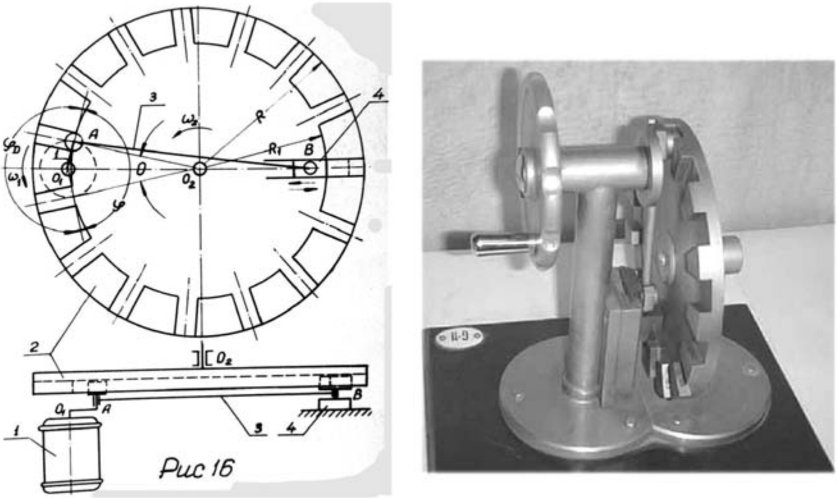


Fig. 3.125: A multi-position Geneva cross-controller mechanism of an electric locomotive

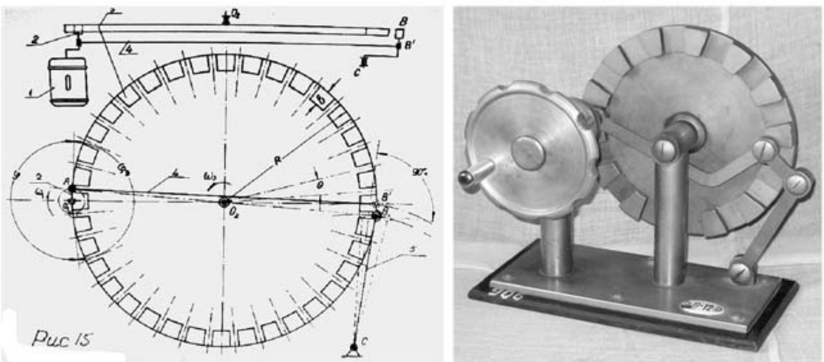


Fig. 3.126: A multi-position Geneva cross-mechanism of a joint-lever lock controller

A few models in the collection belong to the mechanisms of automatic controllers with an electro pneumatic drive. The control of this drive is performed by electromagnet valves, and the power drive is pneumatic. Reshetov offered to change this drive to a drive with two pneumatic cylinders and cam mechanism. A natural model of this drive (Fig. 3.127) was kept in the collection of mechanisms until 1970. Currently, this model is now lost. The scheme of the mechanism from the dissertation of Pavlenko [93] is shown next to the photo in Fig. 3.127.

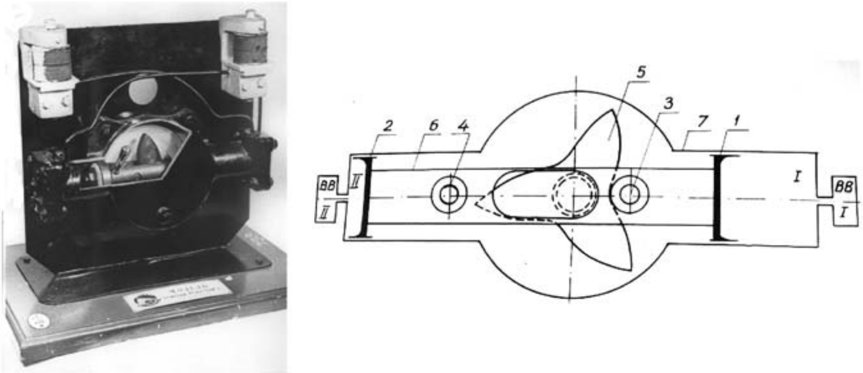


Fig. 3.127: An electro pneumatic two-cylinder cam drive of a controller

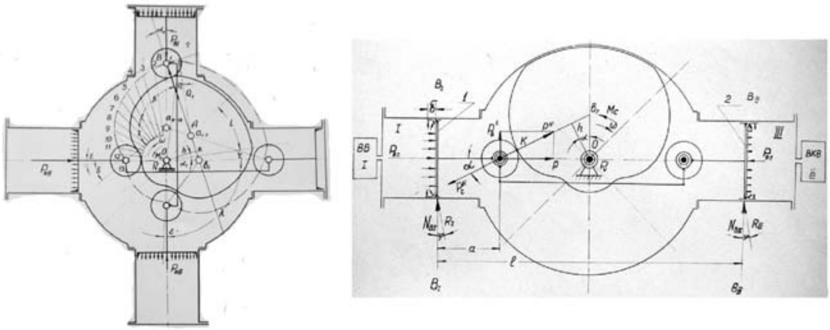


Fig. 3.128: Schemes of mechanisms of controller drives with four and two-pneumonic cylinder cam mechanisms

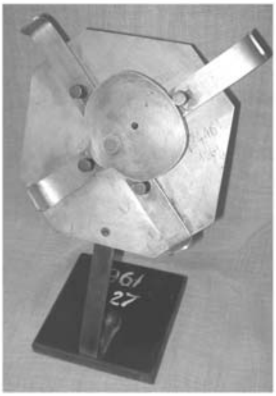


Fig. 3.129: The model of a controller with a cam mechanism with four pneumatic cylinders

One more drive with a cam mechanism was developed by L. Reshetov in the co-authorship with N. Pavlenko. It has two constructive modifications: with two and four pneumatic cylinders. The schemes of these mechanisms [93] are given in Fig. 3.128. There are two models of this mechanism in the collection of TMM department. A photo of one of these models is pictured in Fig. 3.129.

The model shown in Fig. 3.130 represents the level-ratchet mechanism of remote control by a speed regulator of Brown-Bovery's turbine [30]. This mechanism transforms the oscillating movement of the input level into a step rotation of the output link that is the gear wheel with rectangle profiles of teeth. When the control level moves down from the central position the tooth of the ratchet mechanism rotates the gear wheel in one step, clockwise. If the level is moved up from the central position, the gear wheel will get the angle movement in one step but in the opposited direction – counterclockwise. The level is hung in the central position by springs.

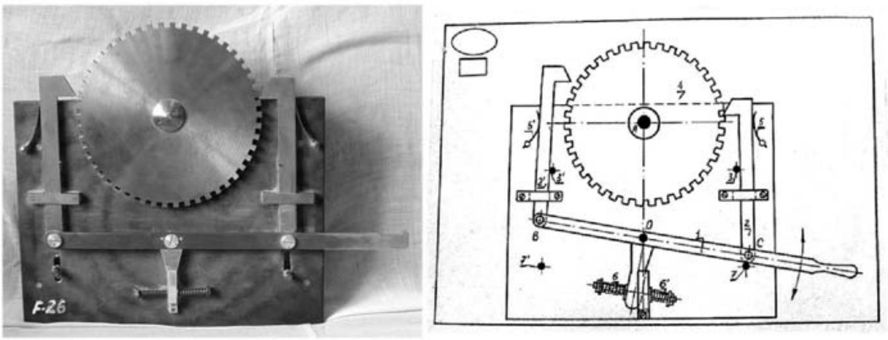


Fig. 3.130: The model and scheme of the reversible lever-and-ratchet mechanism of Brown-Bovery's turbine control

Fig. 3.131: The model of a trundle-cam mechanism of a controller

There is one more mechanism in the collection (Fig. 3.131) which is supposedly connected to the work [93]. There is neither a scheme of this mechanism nor mention of it in the work. However, a likeness to its construction with mechanisms shown in Figs. 3.125 and 3.126 allows us to consider that this mechanism can be connected to the controller's mechanisms. The model consists of a trundle-cam mechanism performing then mechanism's functions of rotation of a temporary shaft (at 180° for one step of the output shaft's movement) and the mechanism of its fixing (on a greater part of the input shaft's revolution) and end-face trundle gearing. In this gearing, the input trundle cog-wheel has two trundles and the output one has 32 end-face mortices (or teeth). During one revolution of the input shaft, the output shaft swings 1/32 of a revolution. This is the principle of the mechanism's work: when one of the right flange's trundles comes into contact with the cam's tooth, it starts to rotate this shaft. At this time, the lock's trundle hits the cam's mortice and allows the temporary shaft to rotate. The shaft rotates at 180º and moves the output wheel one step. At the end of this movement the trundle of lock comes into contact with the cylindrical shape of the input shaft's disk. The temporary shaft is fixed when the lock's trundle slides on the cylindrical shape of the disk. This mechanism performs the rotation of the output link with great acceleration that causes significant dynamic stress. Therefore, using these mechanisms is possible at low speeds only. The models in Figs. 3.122, 3.124–3.127 are similar in constructive design and were possibly planned by one designer. These models were made in the educational workshops of the TMM department in 1960–1965.

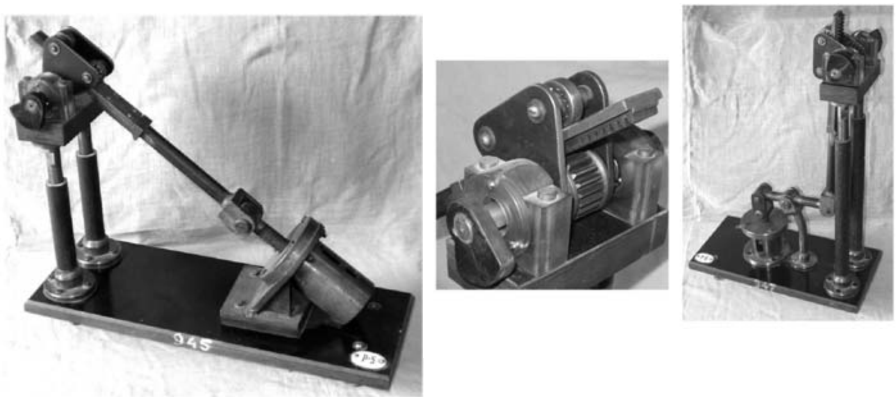


Fig. 3.132: Models of controller mechanisms with a pneumatic drive

Two models in the collection (Fig. 3.132) refer to controllers operated with the help of a pneumatic drive. These models consist of a pneumocylinder on which a rod is placed on the rack of the tooth gearing. Cams of the controller are installed on the shaft of the cog-wheel of this drive. Both models are similar in design and were made during the1950s and 1960s of the last century. The most probable author of these models is L. Reshetov.

Redundant constraints are partially eliminated in the models. For this purpose, in the left mechanism, the rod of the pneumocylinder is executed in the form of two links connected by a rotary pair, and spherical bearings are used in a rack support.

3.7.2. Pantographs

Reshetov and his pupils have made a great contribution to the improvement of pantograph mechanisms for electric locomotives. The dissertations [100, 101] on this theme were written and presented by his postgraduates V. Solomin and S. Aliev. The pantograph is a complex spatial linkage. Provision of the minimal change of pressure upon a contact wire at a change of height is one of the basic requirements of this mechanism. The friction considerably increases in the joints of the pantograph due to manufacturing tolerances, assembly misalignments and temperature differences. This influence can be lowered essentially if the pantograph's mechanism is built without redundant constraints. Some designs of such pantographs have been developed by L. Reshetov in collaboration with his post-graduate V. Solomin. Two models of pantographs are represented in Fig. 3.133: on the right the symmetric and on the left the asymmetric mechanisms are shown.

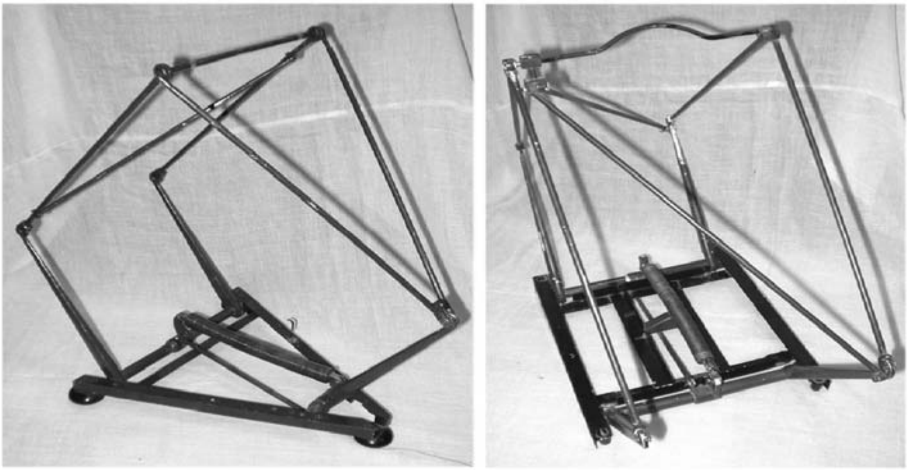


Fig. 3.133: The mechanisms of pantographs without redundant constraints

3.8. Mechanisms of the Kazan's scientific school

3.8.1. Original toothing (Shitikov – Bajazitov gearing)

TMM's Kazan's school was established by Professors B. Shitikov and P. Mudrov. The works of its representatives on equilibration and balancing, spatial linkages, and tooth gearings are known. Scientists of Kazan always kept in touch with TMM department of BMSTU. They gave scientific reports at the department, defended their thesis at the BMSTU academic council, published their works in BMSTU, raised the level of their skill at BMSTU. These contacts also were reflected in the collection of mechanisms.

Some models were presented to the department by Kazan's scientists. These models were: rim gearing (Shitikov – Bayazitov's gearing) and Bennet – Mudrov's spatial linkages.
Rim gearing relates to a new kind of gearing. The first mechanisms with such a gearing were developed by B. Shitikov in 1959. Gears of rim gearing have external screw teeth, and the gearing has a positive gear ratio that is its internal gearing. Its output link rotates in the same direction as its input link. The pitch point of this mechanism divides the line of its centers externally. The scheme of internal gearing with an external location of gears is shown in Fig. 3.134. The geometry and kinematics of this gearing was studied in detail in N. Bayazitov's thesis [102].

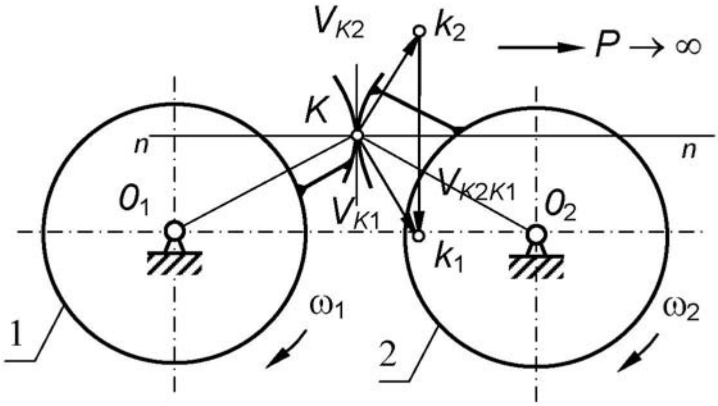


Fig. 3.134: The scheme of internal gearing with external location of gears and the gear ratio u = + 1

Gears both with external and internal locations and with positive and negative gear ratios were received there. Change of sign of the gear ratio is carried out by the appropriate choice of gearing geometrical parameters. Schemes of such gears from article [103] are shown in Fig. 3.135. There is an edge contact of the teeth of wheels between the tooth surface and the tooth points surface of this gearing. Here (Fig. 3.136a) the teeth of wheels can be pointed, with zero thickness of teeth on addendum circle. Teeth having given thickness on this circle are more preferable (Fig. 3.136b).
Having the same direction of teeth screw lines of both wheels is necessary for receiving gearing with a positive gear ratio (internal gearing). Teeth screw lines of wheels have different directions in external gearing (negative gear ratio).
The author developed not only the design procedure of such gears but also the technology of their manufacture. Prototypes of gears were made and tested. Tests showed that gears have low efficiency (close to the efficiency of worm gears). This is caused by high speeds of profile sliding in the gearing. Teeth heating in this gearing is less than in worm gears of identical efficiency. It is defined by design features of the gearing which provide the best heat abstraction from the catching area. The front sections of gears of a rim gearing are shown in Fig. 3.137. The number of teeth of wheels of the gearing are $z_1 = 2$ and $z_2 = 6$, the gear ratio is positive and equals u = 3.

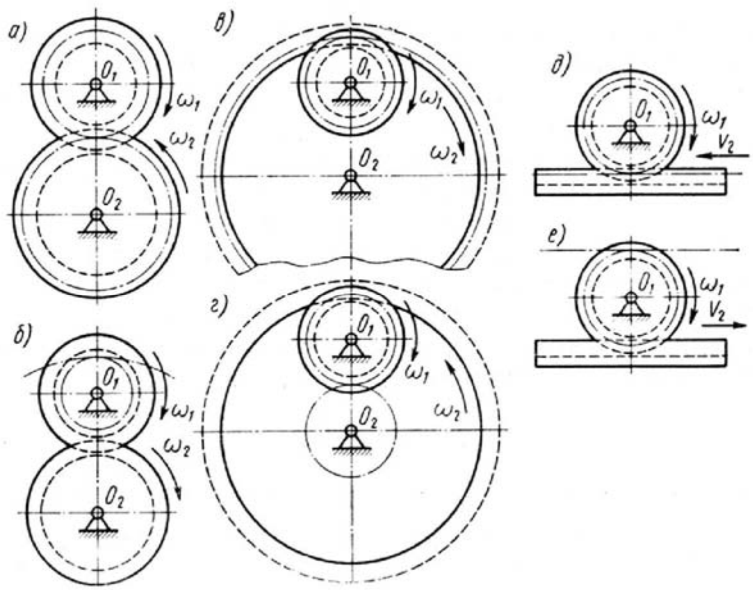


Fig. 3.135: Rim gearings with external and internal location of gears

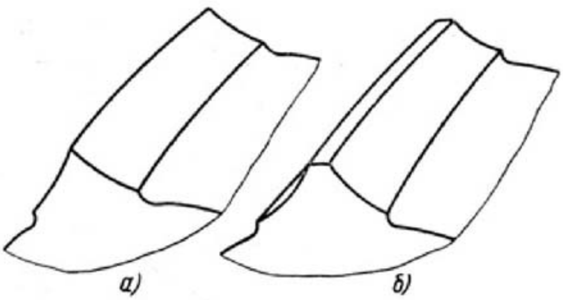


Fig. 3.136: Teeth form modifications of rim gearing

Prototypes of the given gears have found application in locking differentials of crosscountry vehicles and tractors. The differential consists of two pairs of helical gears: a pair of gears with positive (u = 1) and a pair with negative (u = -1) gear ratio. This differential has a very high blocking factor (up to 8). The mechanism of the differential is structurally simple, and possesses small weight and dimensions. The schematic view of design of the differential is shown in Fig. 3.138. The differential has three power streams (three blocks of satellites) for a reduction of loading in the gearing.

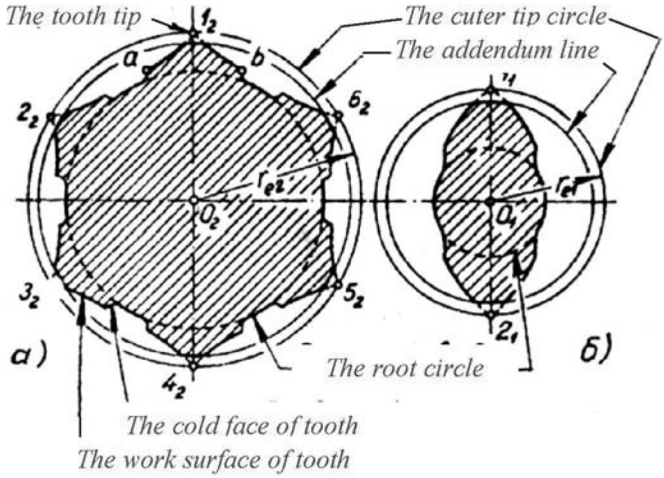


Fig. 3.137: Front sections of wheels

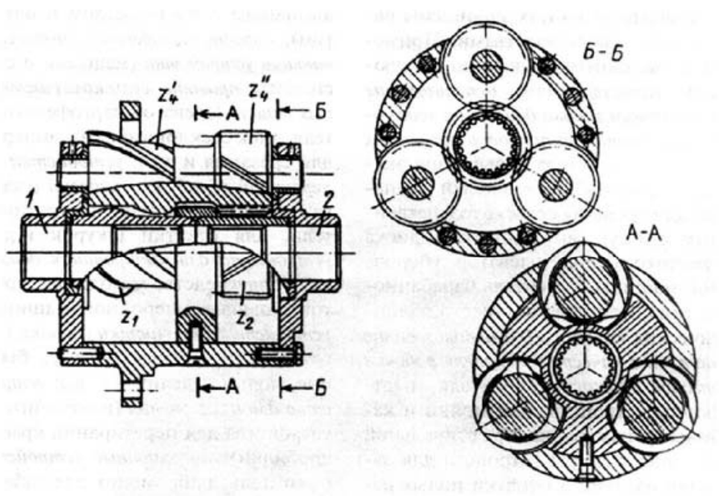


Fig. 3.138: Locking differential of a KamAZ with rim gearing

There is a model of rim gearing in the collection of mechanisms. Gears of this mechanism have external screw teeth with helix angle $\beta = 40^{\circ}$. Both gears have one helix direction (right), teeth numbers of the wheels are $z_1 = z_2 = 6$. The gear ratio of the mechanism is positive $u = +1$. The mechanism represents internal tooth gearing formed by gears with external teeth. A photo of this model is shown in Fig. 3.139.

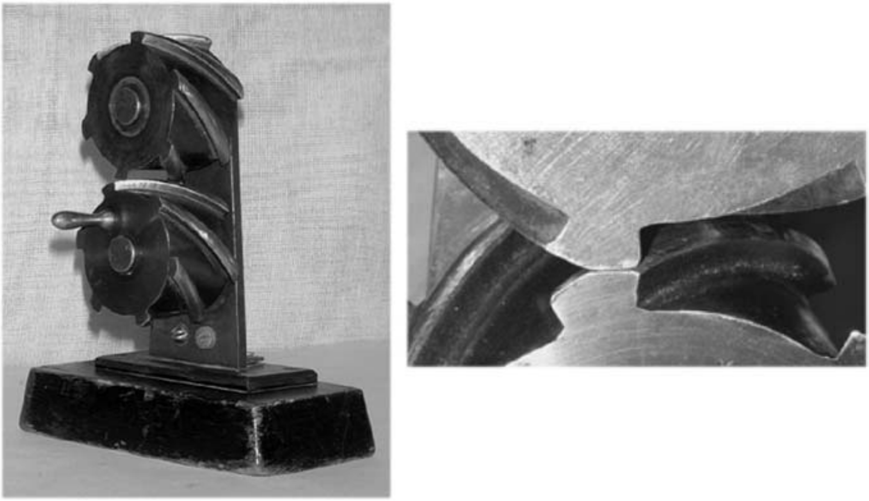


Fig. 3.139: Model of a rim gearing from the BMSTU TMM department collection of mechanisms (on the left is its general view, on the right – its catching area)

3.8.2. Multibar space linkages based on Bennett's mechanism

Spatial mechanisms with rotary pairs are one of Shitikov scientific school's lines of investigation. This line is connected with the works of P. Mudrov and his pupils [104, 105]. The technique of spatial linkage design with set properties was developed by representatives of this school. It is known that formation of such mechanisms is possible only if they consist of seven links. Creation of mechanisms with smaller number of links demands fulfillment of some conditions concerning angular and linear parameters matching.

The method of spatial linkage design based on consecutive association of schemes of known four-bar linkages was developed by Mudrov. Further, it was replenished by the method of division of complex mechanisms. On the basis of these methods, dozens of new schemes of mechanisms have been developed. For these mechanisms a method of kinematics research which allows receiving kinematics characteristics of five-, six-and seven-bar linkages has been created. A method of power analysis of mechanisms with redundant constraints has been offered. Methods of spatial linkages balancing have been developed. Models and prototypes of mechanisms have been created for many schemes of mechanisms. Experimental research of prototypes have been carried out, in particular, efficiency of mechanisms of four-bar linkages η = 0.97–0.98, five-bar linkages η = 0.95–0.96, six-bar linkages η = 0.9–0.92 has been investigated. A manufacturing technique of spatial mechanisms which can be used in individual and small-scale manufacture, based on general-purpose equipment, has been developed. About a hundred copyright certificates and patents have been received on the developed mechanisms. Many mechanisms have been applied in various areas of mechanical engineering, the chemical industry and agriculture. Photos of models of spatial linkages from the BMSTU TMM department collection of models are shown in Figs. 3.140–3.142.

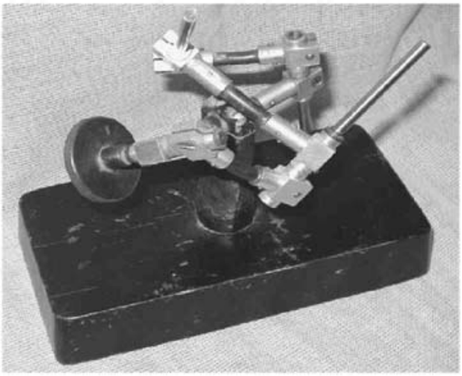


Fig. 3.140: The model of a spatial four-bar linkage

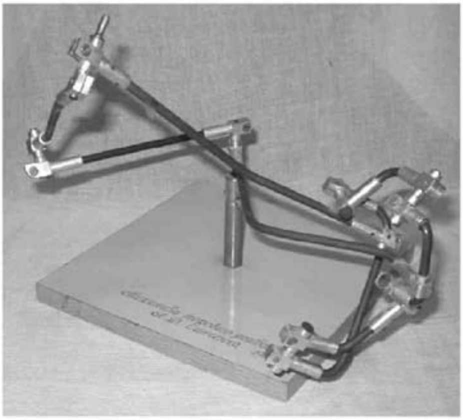


Fig. 3.141: The model of a spatial eight-bar linkage (by L. Sinchenko)

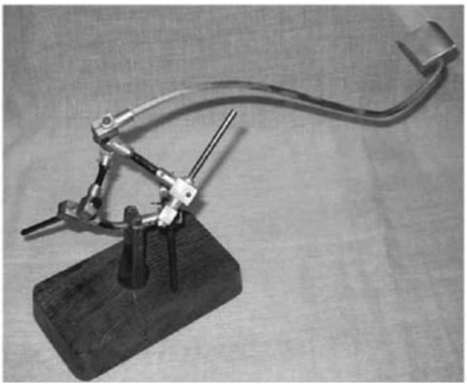


Fig. 3.142: The model of a spatial four-bar linkage

3.9. Models of the Reshetov scientific school

3.9.1. Original gearing

Many sections of the collection are connected with the name of Professor Leonid Reshetov. They are models of kinematic pairs and connection linkages, lever and cam mechanisms, gearings and others. Many of them have been described above and here models which have not been examined in previous chapters, are considered. Models of involute gearings were the first models created by him. In the 1930s, the attention of scientists and designers was attracted to involutes gearings with wheels with small teeth numbers. Creation of such gearings with satisfactory qualitative measures demands the choice of combination of wheel displacement coefficients or a system for correcting gearing. In different countries, different designers and firms developed their own systems of correcting, for example, Kutsbach's (Kutsbach – DIN), Buckingham's (E. Buckingham), Bilgrams' (K. Bilgrams), Schibel's (Ŝchibel) systems and others. These systems were used by various firms producing gearings and reducers, but never published.

The organizing committee of the gear-cutting exhibition held in 1932 suggested analyzing correcting systems and their theoretical justifications to engineer Reshetov. On the basis of researches Reshetov wrote the book [27] in which he not only analyzed existing correcting systems, but also developed his own system. This system allowed the author to develop a gear with the teeth number of wheels equal to 4 (Fig. 3.143). Its helix angle β =11.5º, gear ring width b = 40 mm.

Fig. 3.143: The helical gearing with $z_1 = z_2 = 4$

As a funny thing, Reshetov figured helical gearing with teeth number on the pinion to be equal to one (Fig. 3.144). Axes of wheels of this gearing were parallel. The number of teeth on the wheels was $z_2 = 20$, helix angle – β = 18.3º, gear ring width – b = 40 mm, and diameter of pinion hollows – $d_{f1} = 2.35$ mm.

By Reshetov's calculations "at the laboratory of Krasnoznamenniy (holding The Order of the Red Flag) Mechanic-Machine Building Institute" (the name of BMSTU in 1930) gears were cut which formed the basis for five models of applied mechanics laboratory models. These models are shown in Figs. 3.143–3.146.

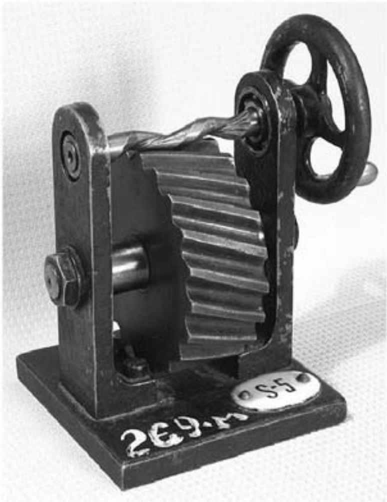


Fig. 3.144: The helical gearing with the number of teeth of the pinion equal to one

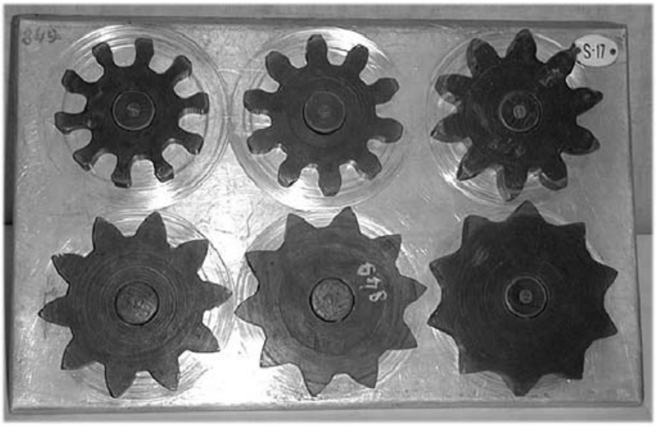


Fig. 3.145: The gears processed with various displacements of the tool

The model in Fig. 3.145 is a set of six gears with equal pitches and teeth numbers but with variable tool displacement – the displacement changes from -0.5 m up to 2 m with step 0.5 m. The model shows the displacement influence on the tooth profile of a wheel. Teeth cutting and pointing is demonstrated there.

Two gearings showed in Fig. 3.146 have equal teeth numbers $z_1 = 4$, $z_2 = 80$ and helix angle $\beta = 15^o$. Differences in gearings of these models are described in [27] in such a way: "Basic advantage of the first model is the greater diameter of the flank circle

(core of the wheel) – 15.98 mm against 13.17 mm. Owing to this, the core flexural strength increases by a factor of 2, which is rather important for such teeth numbers as 4, as it sets a limit on the teeth numbers decreasing. Besides, in the first model, the condition of the smallest wearing is met but is not met in the second one. The only drawback of the first model is the closeness of its teeth to pointing, but it can be easily eliminated. Therefore, the first model has all the advantages over the second one".

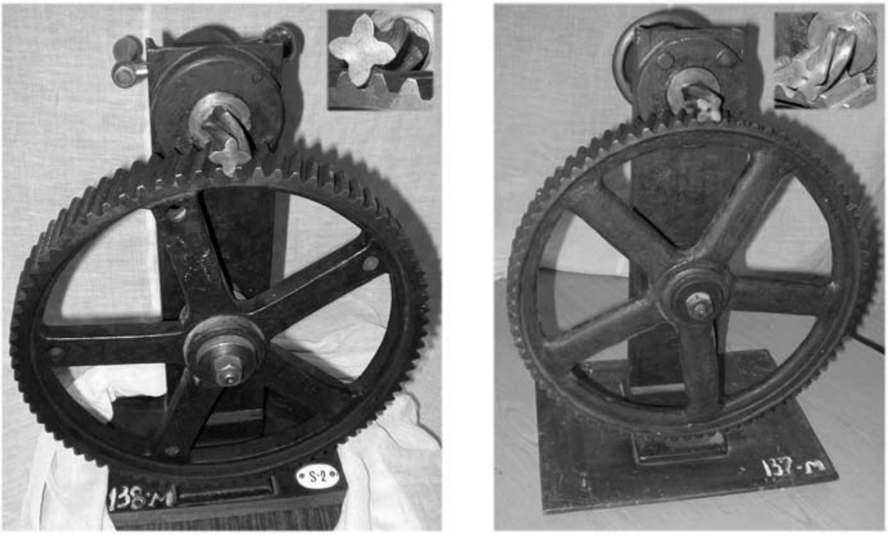


Fig. 3.146: Cylindrical helical involutes tooth gearing with z_1= 4 and z_2 = 80

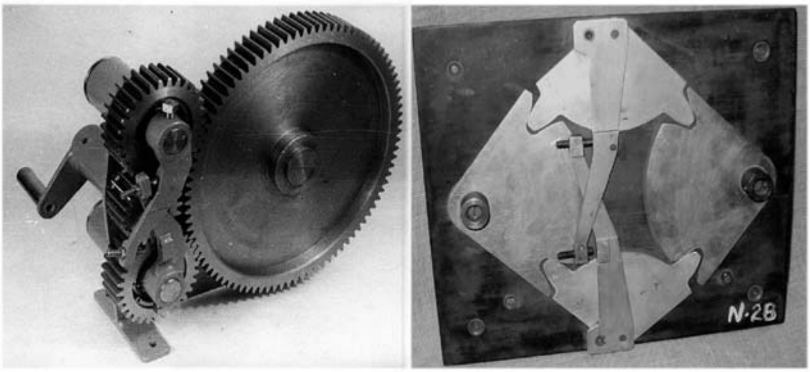


Fig. 3.147: The gear mechanism with the leveling device [104]

Reshetov produced more than 60 inventions and was granted the honorary title of "Honored Inventor of the Russian Federation". He made mock-ups and models of many of his inventions. Some of them are kept in the collection of models. Leonid Reshetov divided these devices into the following categories: electric devices, watch calendars,

Fig. 3.148: Model of the mechanism of an hour calendar

Fig. 3.149: Model of a statically definable support of a cement furnace

gears, traction drives and others. Let us examine the models of these devices kept in the collection. The first two models (Fig. 3.147) represent a traction gear drive of a locomotive with a lever mechanism, which distributes the loading between gears. The mechanism was developed by Reshetov in the codesignership with his post-graduate V. Ermak in 1970. The model shown in the left picture was lost in 1980.

3.9.2. Watch calendar

Much attention was paid to counting device mechanisms in the last century. They were applied in mechanical computing machines, electric, gas and liquid counters, mechanisms of calendars and other devices. Reshetov's nine inventions were devoted to improvement calendars clockwork and models were made of some of them. The important feature of some calendars was fast or "instant" change of date. Some of the calendars were applied in industry, in particular in some models the "Luch" ("Ray of light" – in English) watches of the Minsk watch factory and "Flight" watches of the Moscow watch factory. One of the watch calendar mechanisms is shown in Fig. 3.148.

3.9.3. Statically determinate supports of a cement furnace

Post-graduate I. Gulida researched and developed rational mechanisms of idlers of a cement furnace in his thesis [107]. These idlers are exposed to great loading, therefore the elimination of redundancy is the important scientific and technical problem. The surface which contacts with rim support, is deformed both under the influence of weight of the oven and under the influence of temperature drop. The inaccuracy of installation of both the furnace and its idlers has significant influence on loadings in the idlers. The mechanisms of idlers developed by Gulida allowed the elimination of redundant constraint in the idlers of cement furnaces. A model of such support is kept in the collection of BMSTU. A photo of this model can be seen in Fig. 3.149.

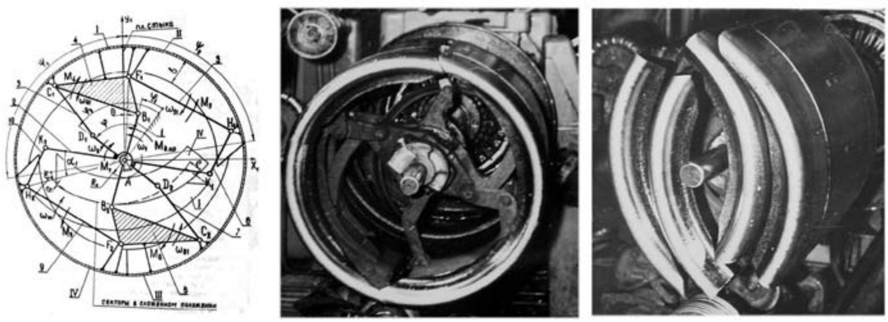


Fig. 3.150: The circuit and photos of skilled designs of developing cord winding drums during the manufacture of automobile tyres

3.9.4. Takeup unit for tyre cord

Ju. Samohvalov's Ph.D. thesis [108] was devoted to the development and research of mechanisms of cord winding drums. These drums are used in trunks of heavy haulers production. After cord winding and ending of the manufacturing process of the trunk, the drum is folded and sent through the trunk side bore. The difference between the sizes of the set-up and folded drum is quite big, therefore creation of designs of such mechanisms is a challenge. Rational design of drums, which did not contain redundant constraints, were developed by Samohvalov. Illustrations from his thesis are represented in the photos shown in Fig. 3.150, and the models created according to the materials of the thesis are shown in Fig. 3.151.

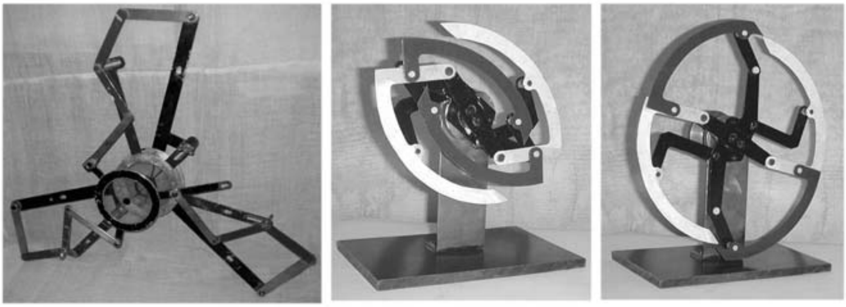


Fig. 3.151: Models of mechanisms of drums for winding a cord during the manufacturing of automobile tyres

3.9.5. Deterioration of the teeth of gearings

Traction gearings of diesel locomotives and electric locomotives belong to mechanisms working in heavy conditions. These gearings work under high unit loads, with lack of lubrication and at significant assembly errors. Teeth of the gearings have significant abrasive deterioration, and are subject to plastic deformations. Two models of the collection represent fragments of worn-out gearings (Fig. 3.152).

On teeth working surfaces of wheels, traces of deterioration and plastic deformations are visible. The form of the worn-out surface shows, real contact surface.

In the bottom part of Fig. 3.152, the scheme from paper [122], showing different kinds of wearing of contacting surface teeth are shown:

- b – pinion profile wearing by bearing
- c – wheel profile galling and appearing of special point of the wheel profile
- d, e – plastic yielding near the pitch point
- f – further pinion wearing by galling

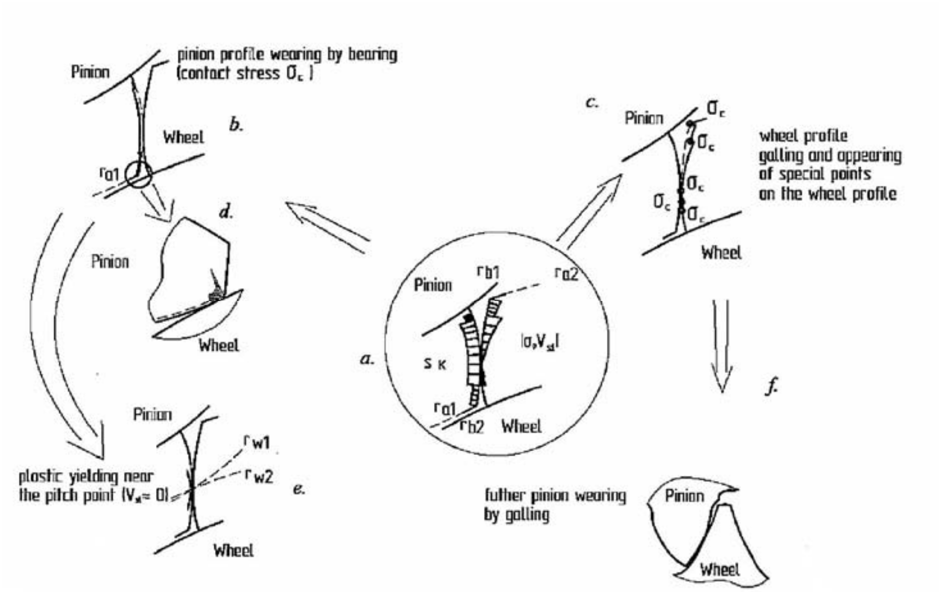


Fig. 3.152: The models of gearings showing deterioration of teeth [122]

3.10. Models of the Gavrilenko scientific school

3.10.1. Tooth gearings processed by standard tools

Professor V. Gavrilenko worked on the geometrical theory of involutes gearings from the beginning of the 1940s. He defended his thesis for a Doctor's degree on this theme in1949 and wrote the monograph [109]. Over almost 30 years, experimental and scientific research in the field of gearings was carried out under his leadership. Some

doctoral theses and more than 20 Ph.D. theses were written and defended and a school of though of involutes gearings was formed. Professors Ya. Davidov, I. Bolotovsky, E. Vulgakov, B. Kurlov, V. Kuzlyakina, H. Kazihanov, K. Sholanov, and G. Timofeev are members of this school. Totally following his disciples they form a big research team which covered the whole of the USSR. This team made a lot of original gearings, many are on the level of inventions. These works found their place in the collection of mechanisms. Many of these models are described in the parts devoted to simple and complicated gearings.

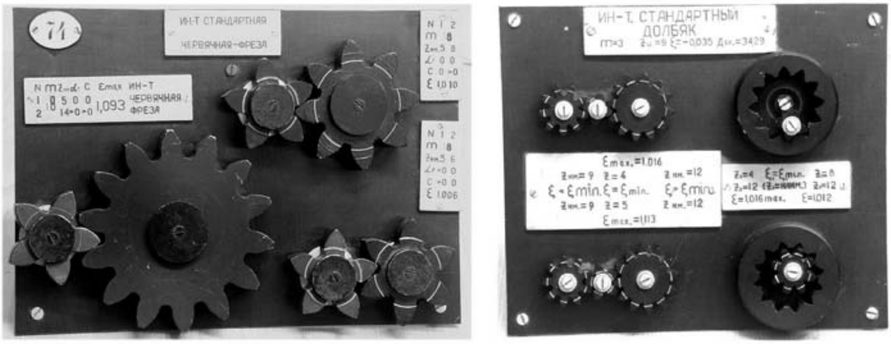


Fig. 3.153: Models of involutes gearings with a minimal number of teeth

Fig. 3.154: Model of external gearings with a low number of teeth

I. Bolotovsky was occupied with research into the geometry of involutes gearings He and his disciples created albums with limiting contours for different types of involutes gearings [110]. Tracing his contours Bolotovsky followed the theory of Gavrilenko. He based on machine-tool action and used standard instrument, as a basis for calculation. Based on research into limiting contours, special kinds of gearings were developed: gearings with a minimal number of teeth [111], gearings with arched teeth [112], gearings with asymmetrical teeth [113], internally toothed gearing with low

difference of a wheel's teeth [114] and others. Models of straight teeth gearings with a minimal number of teeth with internal and external toothing are presented in Fig. 3.154. Gearings with ram tooled toothed wheels are shown in the right photo. The gear wheels in the photo are tooled with spur rack. Initial contour's used the standard parameters. Three external gearings presented in the left photo have such numbers of teeth $z_1/z_2 = 5/14$, $z_1/z_2 = 5/8$ и $z_1/z_2 = 5/6$. Pitch of the gear wheels is m = 8 mm. The gear teeth are sharp. Contact ratio is somewhat larger than one. On the right photo four external gearings ($z_1/z_2 = 4/9$, $z_1/z_2 = 4/12$, $z_1/z_2 = 5/9$, $z_1/z_2 = 5/12$) and two internal gearings ($z_1/z_2 = 4/12$, $z_1/z_2 = 8/12$) are presented. All wheels are ram tooled with the number of teeth $z_0 = 9$ and module m = 3 mm.

The model presented on Fig. 3.154 belongs to this toothed mechanism group. Two involute gearings with the number of teeth $z_1 = z_2 = 6$ and $z_1 = z_2 = 5$ are on this photo. The construction models in Figs. 3.153 and 3.154 belong to the same author and are made from similar materials.

The model created by L. Nikolaev in 2000 is presented in Fig. 3.155. This is a model of a toothed muff with an internal involutes action. The model's wheels have a low number of teeth $z_2 = z_1 = 3$ which is not usual for internal toothing.

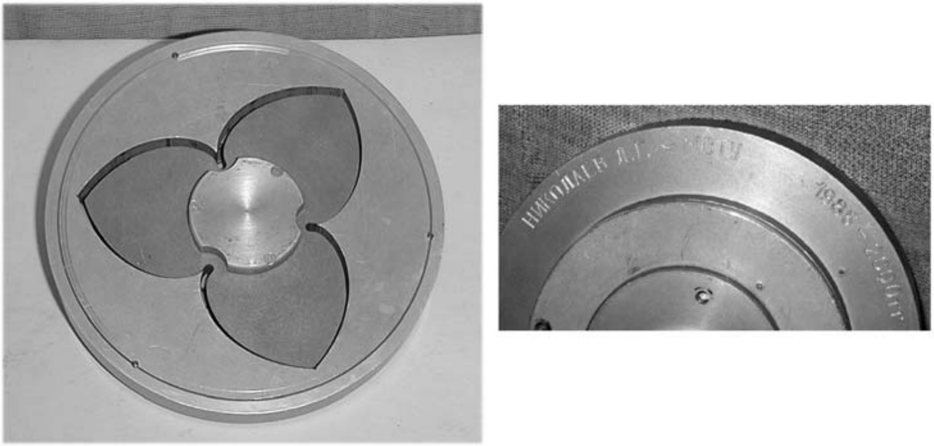
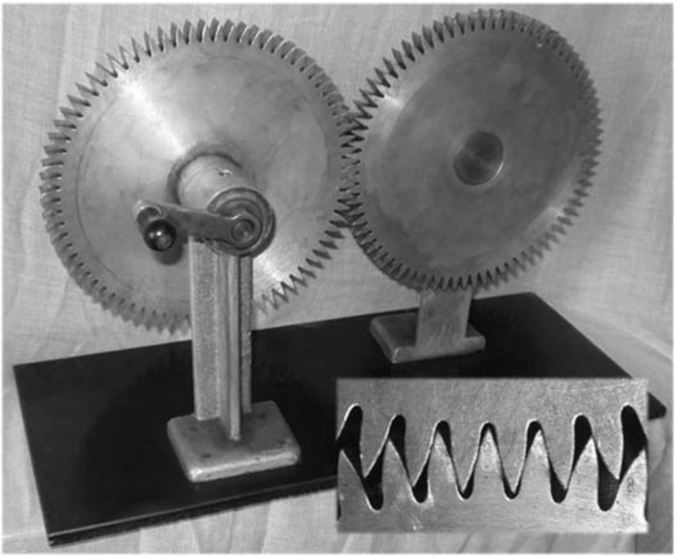


Fig. 3.155: Toothed muff with internal involute engagement and number of teeth z_2=z_1=3

3.10.2. Tooth gearings processed by non-standard tools

Professor E. Vulgakov headed another branch in involutes gearing geometry [7, 8]. The basis of this method of gear-geometry forming is making a gear this needs a qualitative index. First the gearing is created then the tool for clearing a toothed wheel is projected. In that case, the gearing is projected in generalized parameters without recourse to pitch and pitch circle. The use of a special tool allows making gearings with higher qualitative indexes. For example, the model of gearing with wheels tooled with a non standard initial contour has a contact ratio of more than four (Fig. 3.156).

The wheel's teeth have increased height and decreased thickness. Such teeth making up the pitch error and provide uniformity of load distribution.
Experimental research shows that gearings with a contact ratio of two have improved the dynamic characteristic, decreased noise level and work evenness. Designing and producing the special tool are technologically complicated and expensive processes.



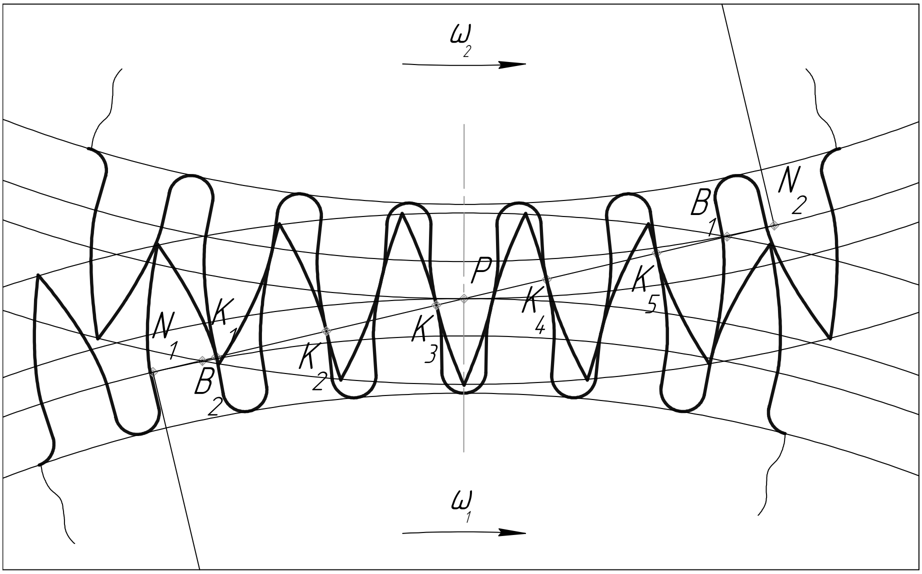


Fig. 3.156: Cylindrical involutes straight toothed gearing with a contact ratio of more than four

Therefore such a method could be used only in special cases and then the weight reduction authorizes the production expense. Involute gearing design styles developed by E. Vulgakov and his disciples allowed mechanisms with some specific properties.

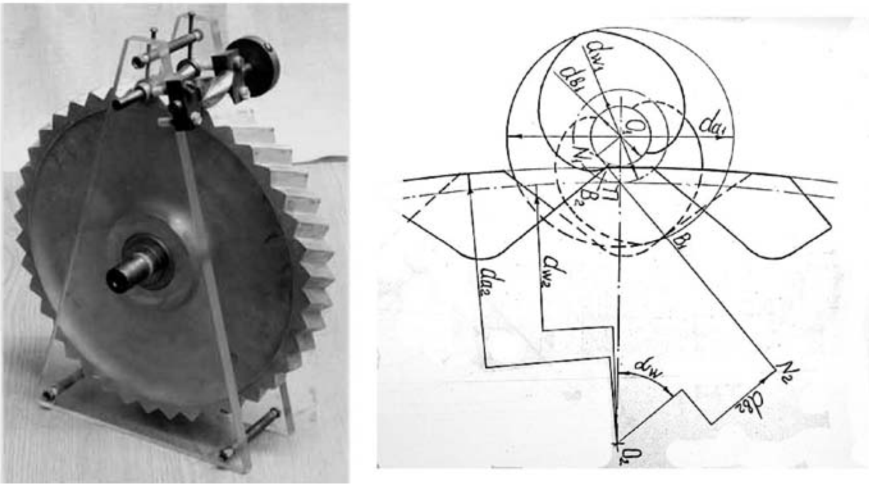


Fig. 3.157: Cylindrical involute gearing with the possibility of one direction of the pinion's turn

The model of oblique tooth gearing with two pinions (Fig. 3.157) allows the irreversible motion. The pinion can transmit motion only in one direction. The mechanism jams if the pinion turns in the other direction. Jamming is provided by a second pinion. Gearing is oblique tooth. Wheels are cut with a large pressure angle. The number of pinion teeth is $z_1 = 1$, the wheel's number of teeth is $z_2 = 40$, tooth angle of slope is $\beta = 13.3°$ End face contact ratio is $\varepsilon_\alpha = 0.66$, the axled one is $\varepsilon_\alpha = 0.59$.

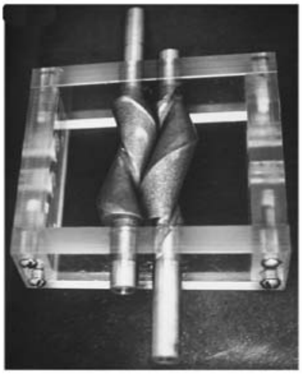


Fig. 3.158: Model of cylindrical oblique toothed gearing with numbers of teeth on the wheels $z_2 = z_1 = 2$

The gearing was projected by postgraduate A. Kapelevich. This is one more model of cylindrical involute gearing with one tooth pinion. The first one was made by engineer L. Reshetov in 1932 [27]. The photo of the model with the number of teeth on the wheel $z_2 = z_1 = 2$ (Fig. 3.158) is presented in Kapelevich's thesis [117]. Such gearings have the peculiarity of toothed wheels: they are made by a special tool instead of a standard tooth-cutting tool. This model was kept in the TMM department collection in the 1970s. Now it is lost.
The thesis of U. Anikin was devoted to researching the geometry of cylindrical gearing with a sine profile (Fig.3.159). A set of toothed wheels with a sine tooth profile and hobbing cutter for cutting such wheels are kept in the collection of the department [118].

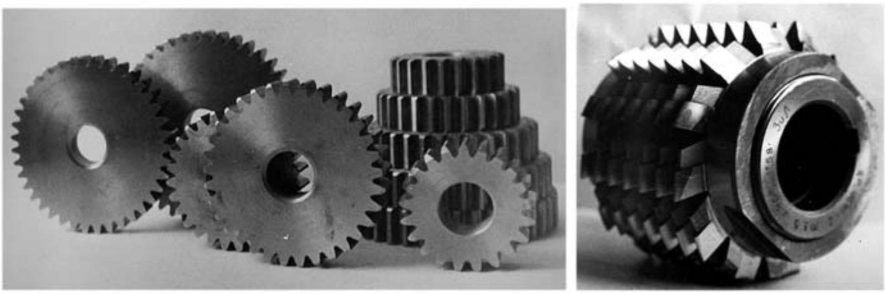


Fig. 3.159: Toothed wheels with a sine profile and a worm cutter for machining such wheels

3.10.3. Tooth gearings of car steering gears

The works of N. Skvortsova and her disciples are described above in Section 3.4.2. However, works devoted to researching the mechanisms of the steering gear of cars are not included in this part. There are three models of such mechanisms in the collection: two with concave worm-roller gearing and one consist of ball-screw gearing and toothed rack gearing. Steering gear requirements for passenger cars and trucks are different [119]. Steering where the reduction ratio is maximal when the steering-wheel position is in the middle, are used for passenger cars In that case the safety improves because when moving at a high speed, a small turn of the steering-wheel doesn't cause a considerable turn of the operated wheels. Besides the steering-wheel's efforts decrease at the beginning of a turn. It compensate for the influences of gyroscopic moment at a high speed.
Trucks usually move at a lesser speeds and possess a greater mass. That's why the reduction ratio of the steering gear must be minimal at low angles of rotation. The total reduction ratio of a truck's steering gear is bigger than that of car. Increasing the reduction ratio when the wheel's angle of turn is increased is necessary for providing ease control. Besides, steering gears must :

- Be reversible in order for stabilization of operated wheels
- Have high efficiency in the forward direction for providing ease of control and low efficiency in the reverse direction for smoothing the shock load reproduced on the steering gear from road asperity

- Provide the necessary rule of reduction ratio changing
- Provide gearing clearance adjustment

Fig. 3.160: Models of steering gears with globoidal worm-roller engagement

Depending on the kind of roller, this mechanism can provide a maximal reduction ratio as well as minimal when moving forward. Both models presented in Fig. 3.160 have a double-comb roller.

Screw steering mechanisms are divided into four types: (1) screw-lever-screw nut, (2) swinging screw-screw nut, (3) screw-swinging screw nut, (4) screw-screw nut-sector. Screw mechanisms can be with a constant reduction ratio as well as with a variable one. There is one model of a screw steering mechanism which consists of a screw, ball nut with toothed cleat and toothed sector in the collection (Fig. 3.161). Application of the ball nut increases the efficiency of the mechanism. The ball nut gap is non-regulated. Therefore the gap in the cleat and sector engagement is regulated in such mechanisms. The cleat and sector teeth are cut with variable displacement. The gap in the mechanism is regulated by axled displacement of the sector. Research into the crew steer mechanism was carried out by a postgraduate student of TMM department, V. Romanova [120] (under the guidance of N. Skvortsova).

Fig. 3.161: Screw-ball nut-sector steering gear

3.11. Models developed by special construction departments

In the 1930s a centralized system of manufacturing manuals for schools and higher schools was created in the USSR. In its framework, demonstration models of mechanisms were also produced. Some of the models are kept in the collection of the TMM department. These models often duplicated Reuleaux's or Redtenbacher's mechanisms of the collection. The model of a planetary belt transmission is shown in Fig. 3.162. It was designed and made by "Techuchposobie" enterprise of Leningrad in the middle of the 1930s. A similar model is kept in the store-rooms of the Polytechnical Museum in Moscow.

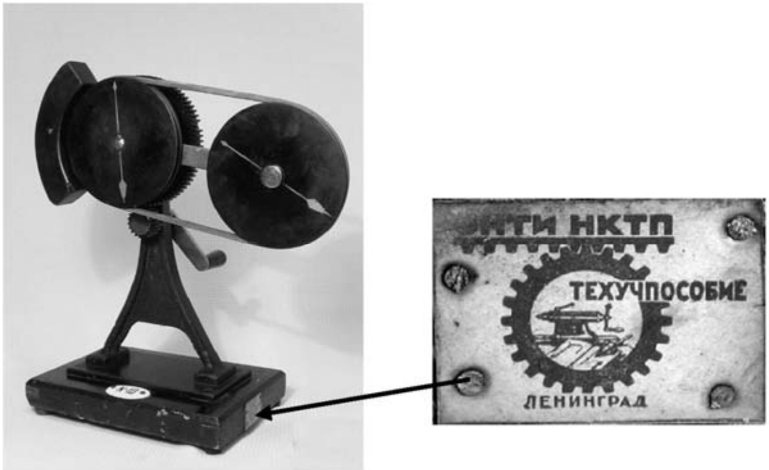


Fig. 3.162: Model of the planetary mechanism with a belt drive, Made by "Techuchposoie" Leningrad

The centralized system of equipping higher schools with manuals worked most productively in 1960s to 1980s. During that time, the SouzVuzPribor association, which was engaged in development of manuals and labware, was created and attached to the Ministry of Higher and Special Education of the USSR. SouzVuzPribor had its own design office and several factories. In the 1960s it created Standard educational TMM laboratories and a collection of demonstration models [121]. Almost all the technical colleges of the USSR were equipped with these laboratory installations and models. Many of them are still used now. Some of these models are available in the collection of the TMM department. Models of cam mechanisms have been described above in Section 3.3. Models of gearing and linkages developed by SouzVuzPribor are shown in Figs. 3.162 and 3.163.

Two models of gearing mechanisms are presented in the first photo: a model of a two-row planetary train and a model of an internal gearing. In the second one there are lever-rocker mechanisms.

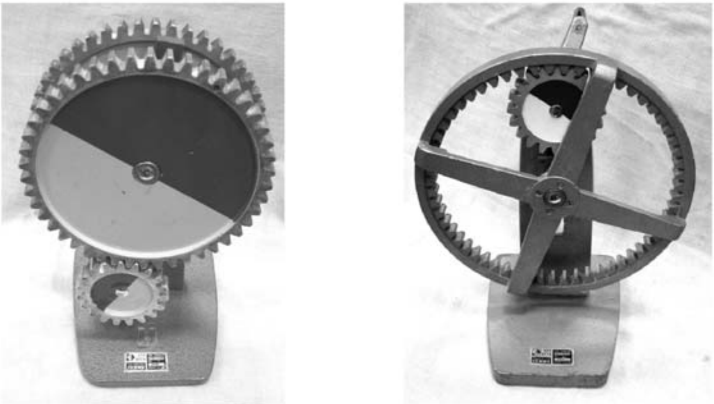


Fig. 3.163: Models of mechanisms by SojuzVuzPribor: a planetary mechanism with external gearings and a tooth gearing with internal gearing

Fig. 3.164: Models of mechanisms by SojuzVuzPribor: linkages

In the middle of the 1980s a new complete set of models of mechanisms and a new standard laboratory of TMM [122] were developed. In comparison with the previous complete set of models, that set had a more simple design, did not contain cast details and had smaller weight and sizes. Models were made according to mechanisms, which were studied by students on the TMM course. The complete set of models included various types of mechanisms: linkages, gear and cam. In the collection of models of the BMSTU only linkages of the set are represented. A part of these models is shown in Fig. 3.165.

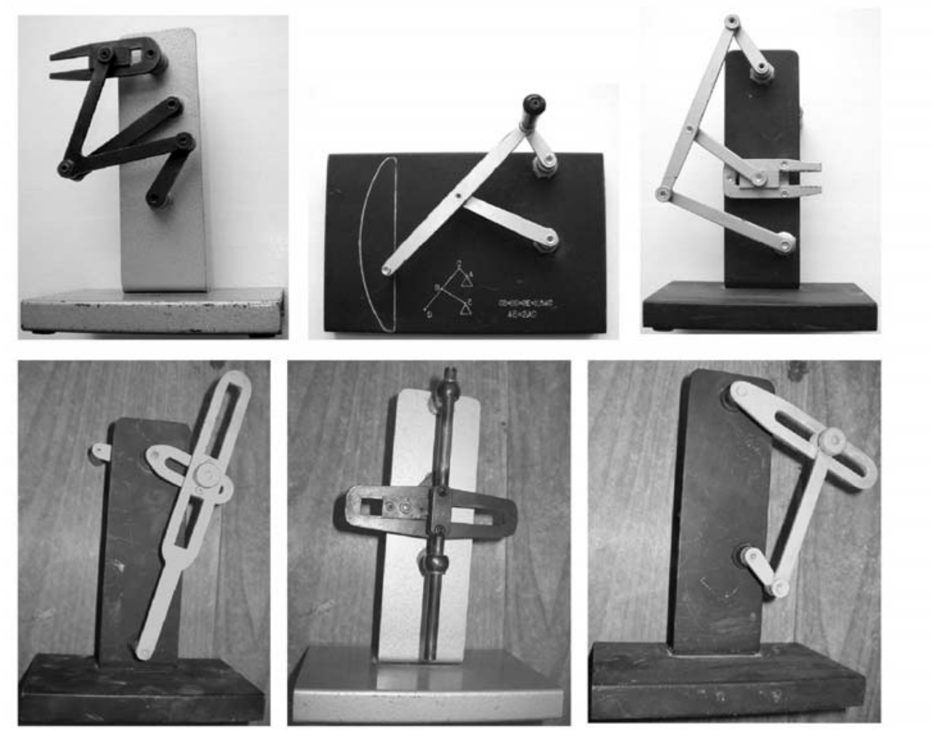


Fig. 3.165: Models of SouzVuzPribor: planar linkages

In 1995, BMSTU's TMM department, in collaboration with RosUchPribor, began creating a new version of the standard educational laboratory of TMM [123]. For this laboratory not only experimental installations, but also models of mechanisms were developed. In laboratory work on metric synthesis of linkages, two models are used: a 4-hinge mechanism and a crank-slide mechanism. The models permit changing the size of their links and making measurements, according to which it is possible to construct the function of the position of the mechanism.
The models were made by the Gagarinsk branch of RosUchPribor. Photos of these models are shown in Fig. 3.166.

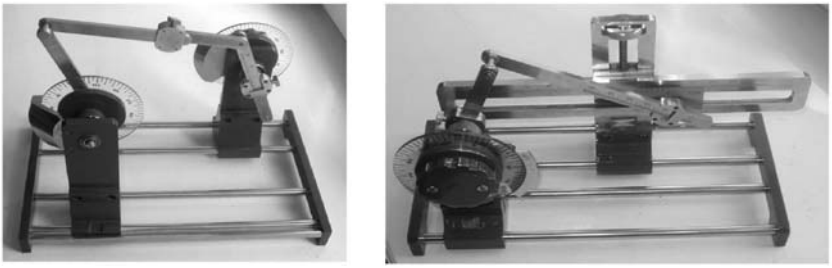


Fig. 3.166: Models of mechanisms for laboratory work on the metric synthesis of linkages

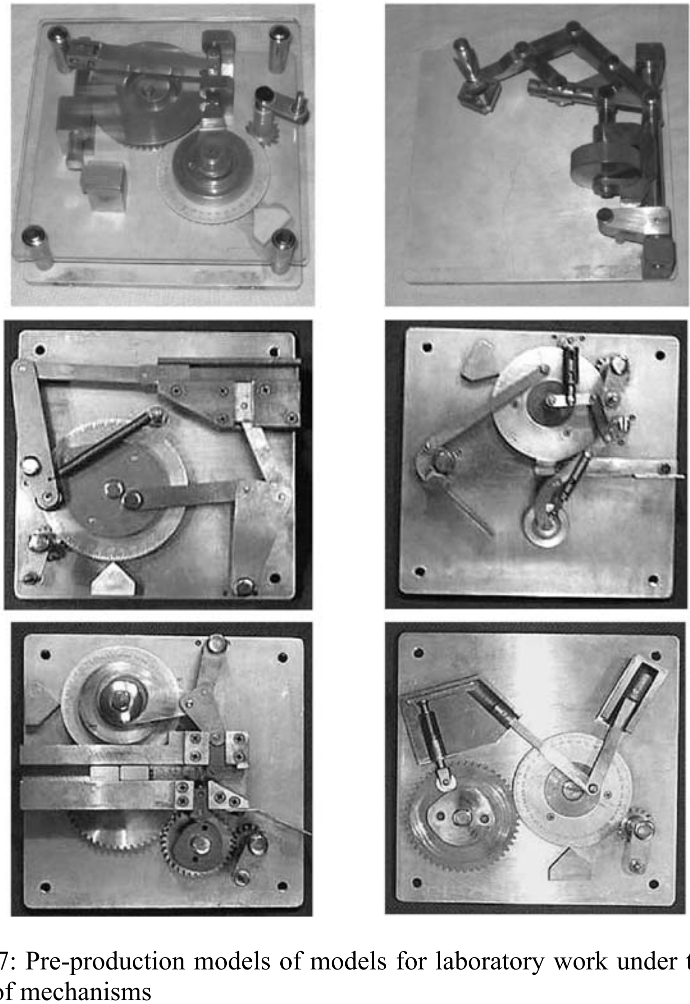


Fig. 3.167: Pre-production models of models for laboratory work under the structural analysis of mechanisms

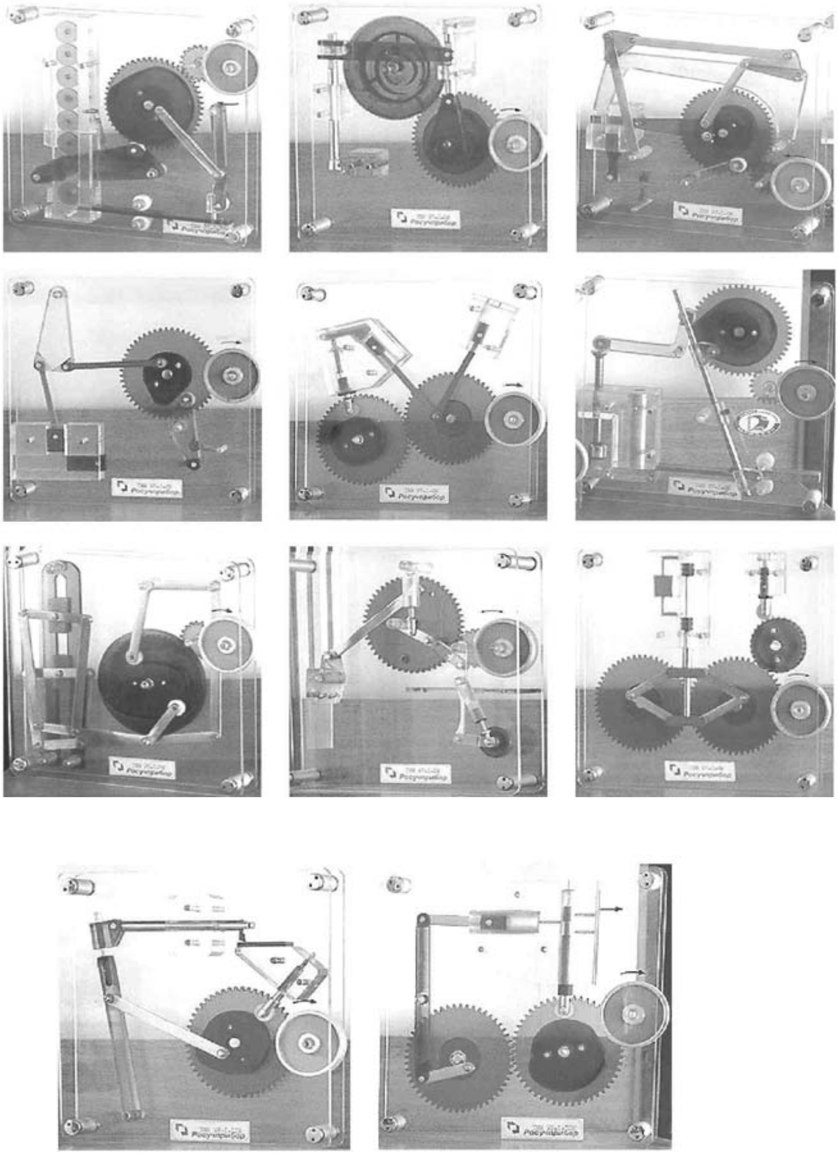


Fig. 3.168: The complete set of models for laboratory work under the structural analysis of mechanisms

Models of mechanisms of various devices were developed for laboratory work on structural synthesis: machine tools, presses, engines, pumps, compressors. The models represent complex mechanical systems which include gearings, cam and lever mechanisms. At the first stages of their development, high durability providing sufficient service life was one of the basic requirements for these models. The first trial complete set was made of metal. The models were heavy, and their appearance was unsatisfactory. Some of these models are shown in Fig. 3.167.

In the final version, basic details of models were made of transparent plexiglass painted in different colors, some of the details were made of metal. The main advantage of transparent models is the opportunity for their demonstration through a projector on a larger scale. So it is possible not only to show a model, but also to show the movements of its parts.

The complete set of laboratory work includes models of the following mechanisms (Fig. 3.168):

- The mechanism of knife frames TMM 97-1-7
- The mechanism of a slotter TMM 97-1-3
- The mechanism of a horizontally-forging machine TMM 97-1-9
- The mechanism of a crank-bend press TMM 97-1-2
- The mechanism of a cross-section-planing machine tool TMM 97-1-11
- The mechanism of an oscillating conveyor TMM 97-1-8
- The mechanism of a combustion engine-compressor installation TMM 97-1-6
- The mechanism of a power ship installation with a Stirling engine TMM 97-1-4
- The mechanism of a front rack of plane chassis TMM 97-1-5
- The blank feeder in a working zone of a machine TMM 97-1-10
- The mechanism of a piston pump TMM 97-1-12

3.12. Models created by the authors of the book

In 1980, the TMM departments of the BMSTU were disbanded and replaced by computer classes. The opportunity both for manufacturing new models and for repair work has disappeared. However, the collection of mechanisms was nevertheless replenished last year. Some models have been made by students of the Kuncevsk branch of BMSTU under the direction of Professor A. Golovin. These models include of the mechanism press [124] which has been described in Section 3.2.1, and a model of tooth gearings – in Section 3.4.3. Outside consideration, there was only a model eccentric a photo of which is given on Fig. 3.169.

Some models of mechanisms have been made by V. Tarabarin: a model pin gear's hydraulic motor and model concerning its mechanism (Section 3.3), a model of a wave tooth gearing (Section 3.4.2), the model rack gearing and a model of the mechanism of a piston compressor. Photos of the two last models are shown in Fig. 3.170. The mechanism of the one-cylinder piston compressor for the model is taken from a domestic refrigerator.

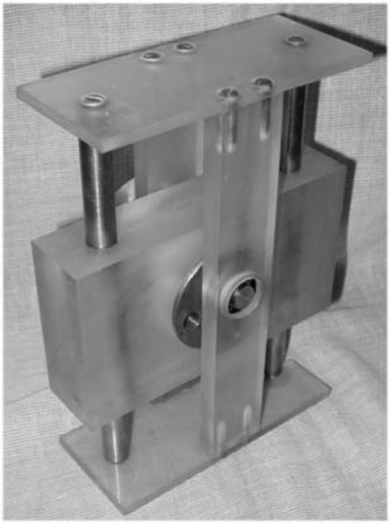


Fig. 3.169: Model of the two eccentric press [125]

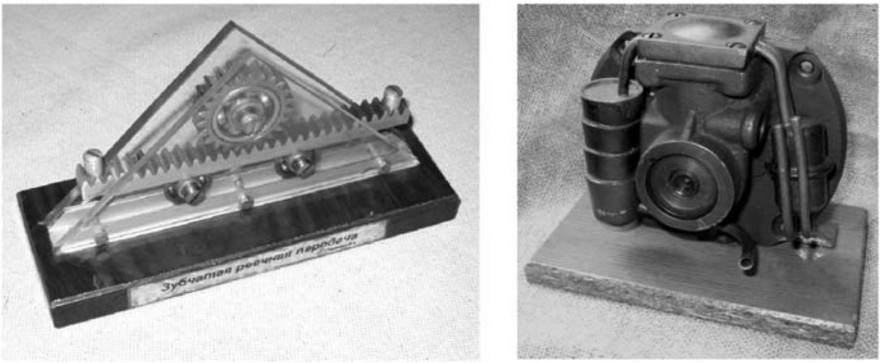


Fig. 3.170: Model rack gearing and model of the piston compressor

APPENDIX

LABORATORY EDUCATIONAL EQUIPMENT USED IN THE TMM DEPARTMENT OF BMSTU IN VARIOUS YEARS

Laboratory work on the definition of the moments of inertia of links

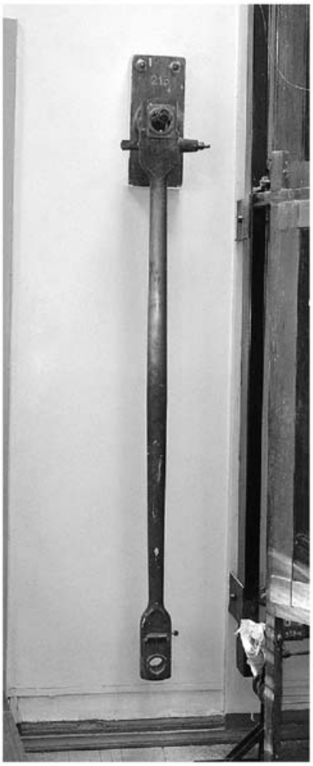
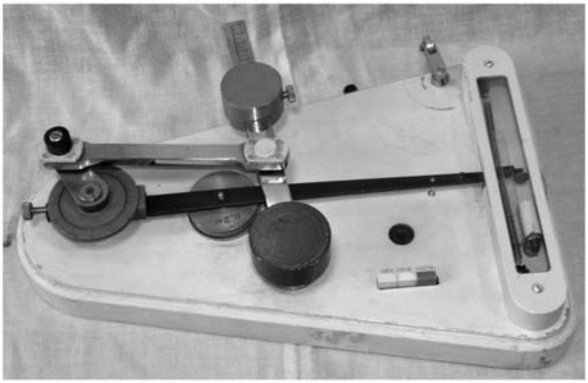


Fig. A.1: Coupler of a steam locomotive which was used by L. Smirnov for carrying out laboratory work on the definition of the moment of inertia during the period of oscillation (about 1930)

Fig. A.2: The testing machine for laboratory work on the definition of the moment of inertia during the period of oscillation (RosUchPribor, 1
980)



Laboratory works on the research the cams

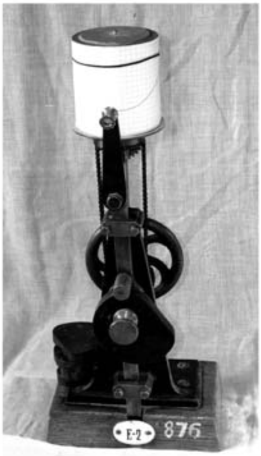


Fig. A.3: The device for researching the kinematics of cams (BMSTU, 1950–1970)

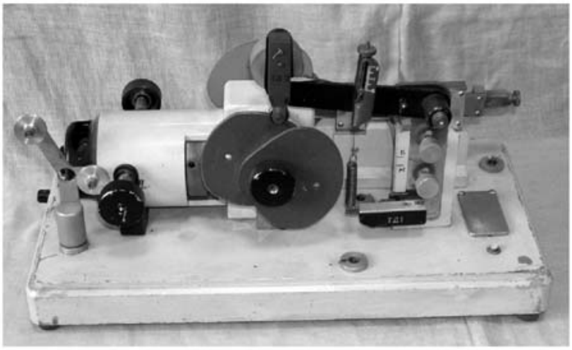


Fig. A.4: Laboratory plant for researching the kinematics and dynamics of cams (RosUchPribor, 1980)

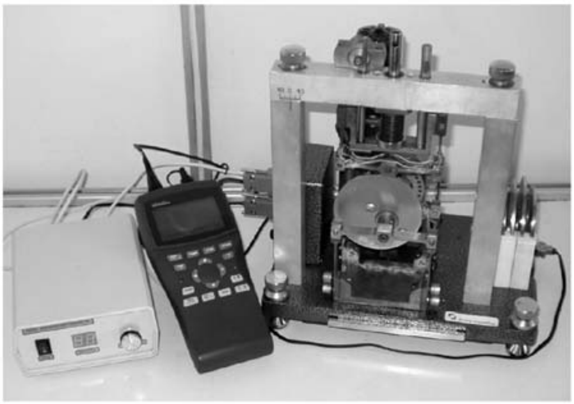


Fig. A.5: Laboratory plant for researching the kinematicsand dynamics of cams (BMSTU-RosUchPribor, 1980)

Laboratory work on the research of a one-cylinder piston compressor

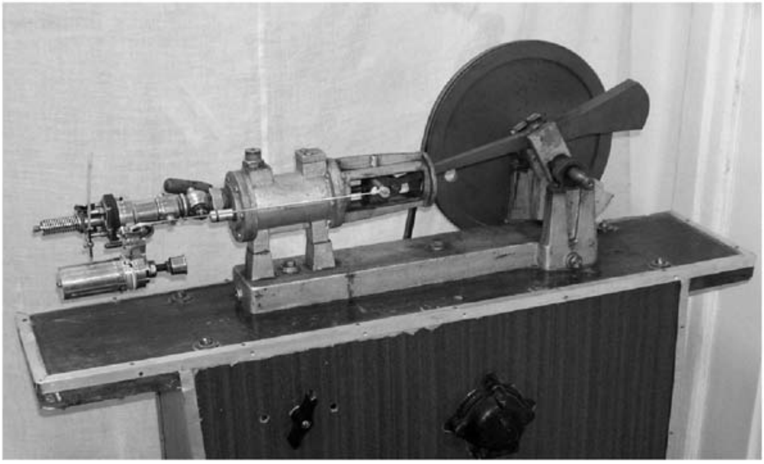


Fig. A.6: The testing machine for researching a one-cylinder piston compressor(BMHTS, 1950)

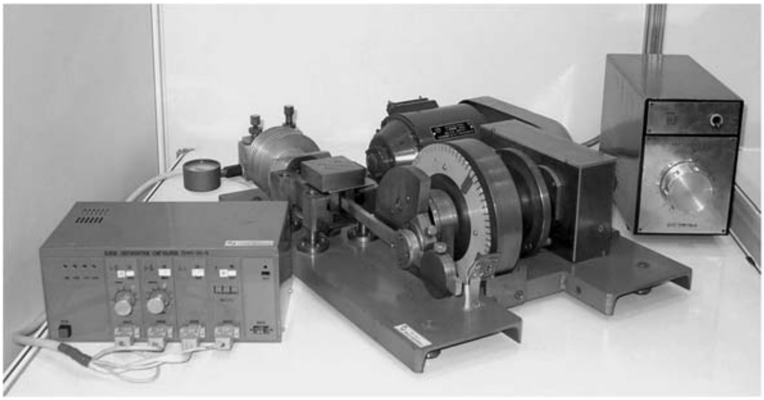


Fig. A.7: The testing machine for researching a one-cylinder piston compressor (BMHTS-RosUchPribor, 2000)

Laboratory works on the profiling a tooth involute's wheels

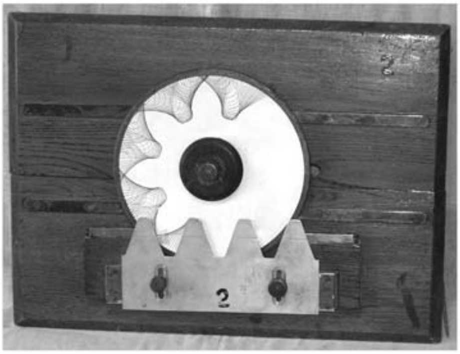


Fig. A.8: Device for profiling a tooth involute's wheels (BMHTS, 1950)

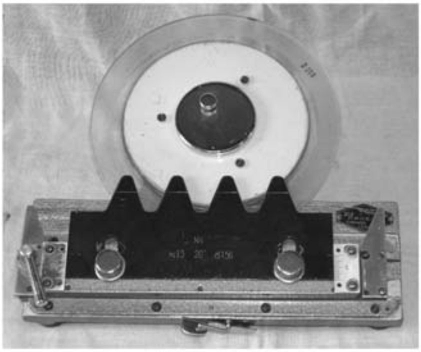


Fig. A.9: Device for profiling a tooth involute's wheels (SojuzVuzPribor, 1960)

Fig. A.10: Device for profiling a tooth involute's wheels (RosUchPribor, 2000)

Laboratory works for defining of the efficiency of a mechanism

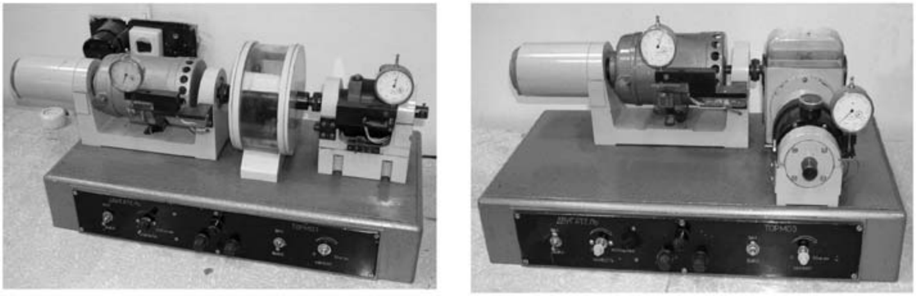


Fig. A.11: The testing machine for the defining of the efficiency of a reducer (SojuzVuzPribor, 1960)

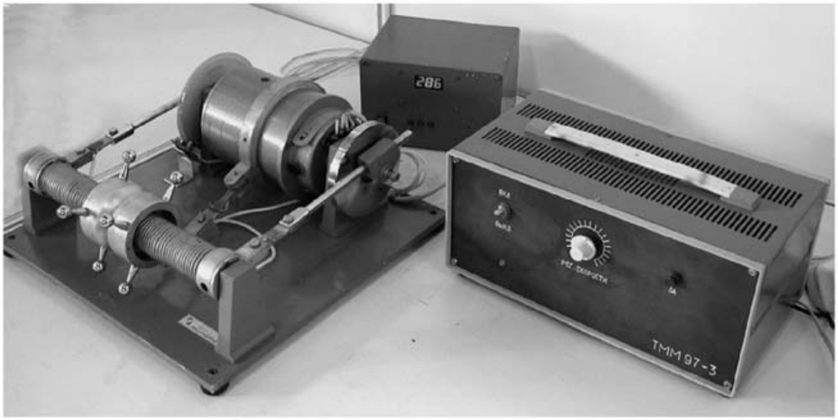


Fig. A.12: The testing machine for the defining of the efficiency of a reducer (RosUchPribor, 2000)

Laboratory work on balancing rotors

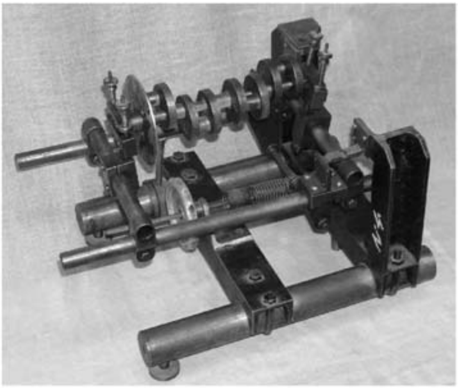


Fig. A.13: The testing machine for laboratory work on balancing rotors (BMHTS, 1950)

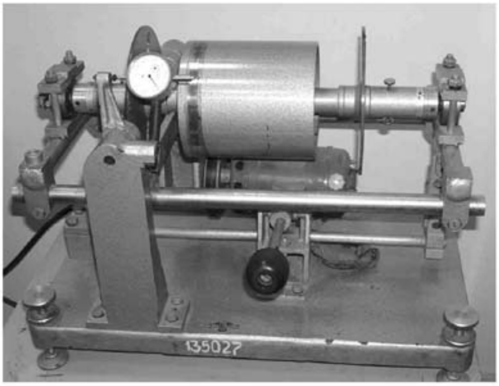


Fig. A.13: The testing machine forbalancing rotors (SojuzVuzPribor, 1960)

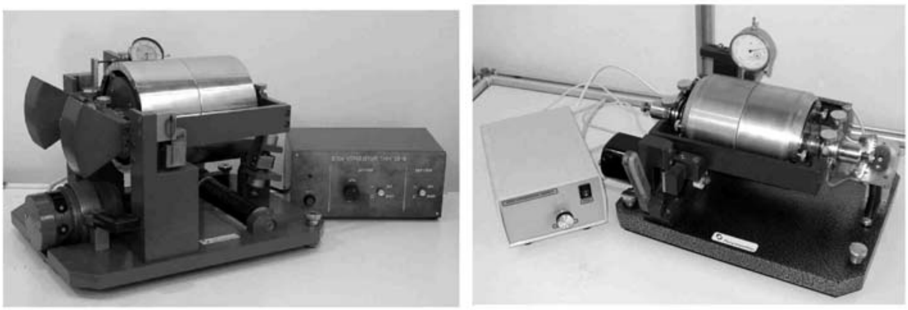


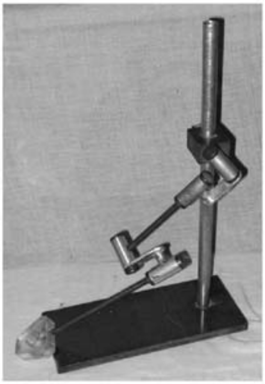
Fig. A.14: The testing machine for balancing rotors (RosUchPribor, 1999 and 2005)

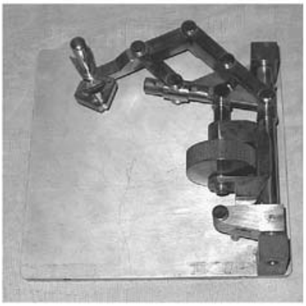
Laboratory work on the research of friction in kinematic pairs

Fig. A.15: The testing machine for laboratory work on the research of friction in prismatic pairs (SojuzVuzPribor, 1960)

Fig. A.16: The testing machine for laboratory work on the research of friction in revolute pairs (SojuzVuzPribor, 1960)

Models and devices for laboratory work on manipulators and robots





(RosUchPribor, 1995)

(RosUchPribor, 2000)

(BMSTU, 1980)

Fig. A.17: Models of manipulators

Fig. A.18: The testing machine for laboratory work on the kinematics and structure of manipulators (BMSTU-RosUchPribor, 1999)

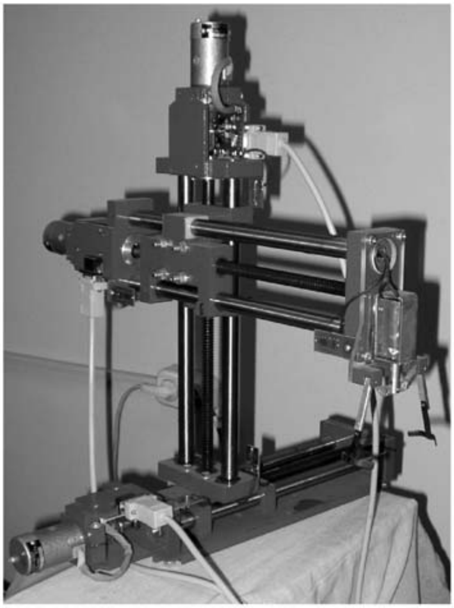


Fig. A.19: The testing machine for l aboratory work on the kinematics and structure of manipulators (MSIEM-RosUchPribor, 1995)

The real industrial robots and manipulators used in BMSTU for execution of laboratory work

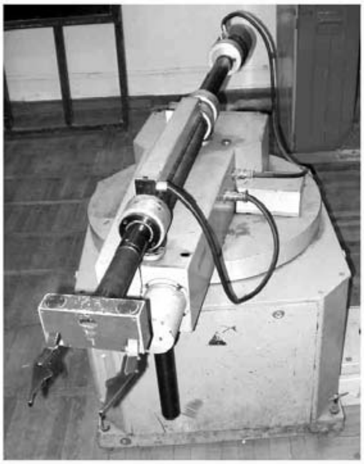


Fig. A.20: Copying electromechanical manipulator MEM-3ñ

Fig. A.21: Electromechanical industrial robot M-901

Fig. A.22: Pneumonic-mechanical industrial robot PRP1-1

The real industrial robots and manipulators used in BMSTU for the execution of laboratory work

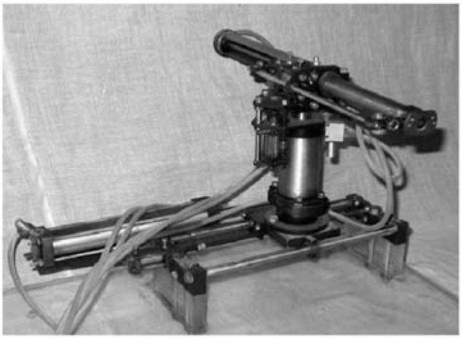


Fig. A.23: Pneumonic-mechanical industrial robot PRP1-1 (rectangular system of coordinates)

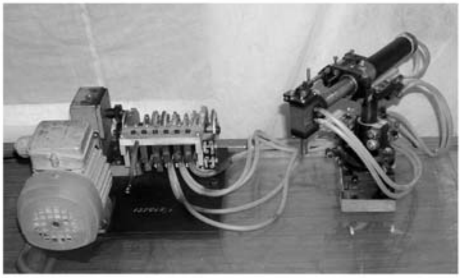


Fig. A.24: Pneumonic-mechanical industrial robot PRP1-1 (cylindrical system of coordinates)

Laboratory work for researching of methods of vibroprotection

Fig. A.25: The testing machine for researching of methods of vibroprotection (RosUchPribor, 1999)

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BIOGRAPHICAL NOTES

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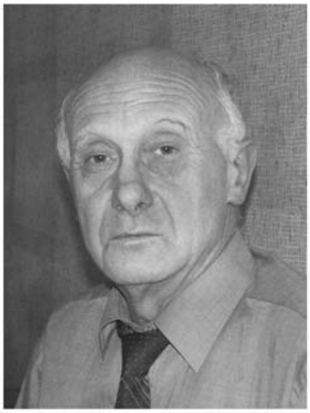

Alexander Golovin has graduated with a degree of mechanical engineer with laude (stamping equipment and technology) in 1962 from Bauman Moscow State Technical University. In 1971 he has graduated with the degree of Candidate of Technical Science (stamping equipment and technology), in 1993 – the degree of D.Sc. (theory of mechanisms and machines, stamping machines). Since 1968 he has been teaching the course "Theory of Mechanisms and Machines" at the BMSTU TMM Department. Since 1977 he became an Assistance Professor and in 1993 he became Professor of TMM. He has created the task-oriented course of mechanisms for the Departments "Technology and Machines for Forging and Stamping Processes". He is a member of the IFToMM Permanent Commission for the History of Machines and Mechanisms Science, a member of the Russian National Committee on TMM. His research interests cover aspects of the Theory of Machines and Mechanisms and Mechanisms Presses. Specific subjects of his interest are Multivariant Designing of Linkages, Mechanical Analogues of Biological Systems, Teaching and History Problems of TMM. He is author or co-author of two monographs, 5 inventions and more than 100 papers, which have been presented at National and International Conferences and magazines. In 2005 he organized and conducted the International Conference on History of Machine and Mechanisms Sciences in Bauman University (Moscow).

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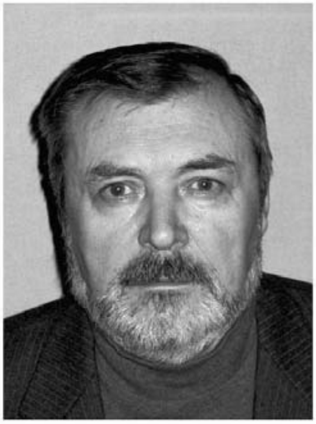

Valentin Tarabarin was born in Moscow in 1944. In 1968 he graduated with the degree of mechanical engineer with laude at the Bauman Moscow State Technical University specializing in "Wheel's and caterpillar's machines". At the time of graduating, he was directed to BMSTU TMM Department as an engineer of Science Research Branch (SRB). He has been engaged in designing and testing mechanical gear drives with harmonic drives and planetary tooth gearings. In 1976 he graduated with the degree of Candidate of Technical Science (differential harmonic drives for servo-control's systems) and continued his activity in SRB. In 1980 he passed on teaching activity. Now he is an Ass. Professor of BMSTU TMM Department and curator of the TMM department collection of mechanism models. He is the head of two Internet projects: TMM educational-methodical complex (http://tmm-umk.bmstu.ru) and the Collection of mechanism models of the TMM department. He is the member of BMSTU Academic council on open education, chairman of the TMM department laboratory commission. Tarabarin is the author of more 100 papers and 10 inventions. Since 2005 he has been a member (observer) and since 2007 – member of the IFToMM Permanent Commission on the History of Mechanisms and Machines Science (MMS).
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- Glossary of Terms “Theory of Mechanisms and Machines”, edited by “Science”, Moscow, 1984. Fascicle 99
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